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Behaviour and movement of locomotive wheels on the track,

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Figs. 1 to 30, pp. 419 to 436.

(*Schweizerische Bauzeitung.*)

In an article entitled: « Rad und Schiene » (Wheel and rail) which appeared in the *Zeitschrift des Vereines deutscher Ingenieure,* Professor Jahn has published the results of trials made with a cylinder which he caused to roll on an inclined plane. These trials have given data which throw light upon the ratio between the peripheral force and the adhesive weight of a wheel, but neglect the forces acting in the direction of the axis of the axle. Jahn's researches deal therefore with the relationship, which has great importance in practice, that exists between the tractive effort and the adhesive weight of a vehicle. But the ratio of the adhesion of a wheel in the axial direction is no less important, because it serves as a basis for obtaining the pressure against the flanges and, consequently, the wear of the wheel and of the rail and the smoothness of running of the vehicle.

Various, more or less contradictory, theories have been propounded relating to this adhesion, but up to the present no known experiments have thrown light on the adhesion in the axial direction at various running speeds for a wheel hauled or braked.

In 1906, von Helmholtz, the well-known railway engineer published in the *Zeitschrift des Vereines deutscher Ingenieure* (1906, p. 1553) an article on the Krauss-Helmholtz bogie, in which he calculates the coefficient of friction for axial displacement (at one-sixth of the load of the wheel). Recently von Helmholtz abandoned this point of view, at least partially; in fact in a report dealing with a new type of bogie for electric locomotives, he makes the following statement:

« Does a rear radial axle find sufficient lateral support to enable it to act satisfactorily as a guide to another vehicle following it on a curve? To this question I had hitherto unhesitatingly replied in the affirmative, but the reply should be in the negative; perhaps if not an absolute negative at least as a

negative as a general rule as shown by the experiments made in 1906. We have to deal with a difficult subject in the complex rolling movement, combined with that of rotation about a centre, with as little friction as possible. It is necessary to make a careful distinction between sudden changes of direction on the one hand, and progressive changes in direction on the other hand. In the article on the *Zeitschrift des Vereines deutscher Ingenieure,* 1906, p. 1554, we considered a sudden change. What we then said may consequently be maintained to-day. But against the disturbing forces that act for a short time only, as also against those which tend to cause swaying motion and are found in running over an undulating track having waves of long pitch (swaying at high speed) or those which act continuously in the transverse direction (centrifugal force or superelevation of the outer rail), the rear radial axle is, between certain limits, devoid of support and must yield to these forces, even though they should be less than the adhesion or merely greater than zero, whenever there is sufficient time for them to come into action.

« The difficulty in the question lies in the fact that the lateral pressure or, which is the same thing, the lateral resistance of an axle is not merely and simply equal to its adhesion A, but equal to $\beta \times A$; where the coefficient β may vary from zero to unity. Under working conditions the leading axles which run at an oblique angle to the rails have a value of β nearly equal to unity. On the other hand in the case of axles that are arranged to lie radially, it is equal to zero in the first instance, and the resistance is only produced as and when the axle is diverted from this radial position by external forces. »

The trials described below have been made with the object of ascertaining the lateral adhesion of the wheel on the rail under the most widely different service conditions; they were carried out on a model which reproduced the conditions encountered in practice as accurately as possible.

For these trials use was made of a lathe (fig. 1) having a horizontal spindle driven by a three-phase motor at a constant speed; the speed was changed by shifting the belt on the step cone. The disk A (fig. 2) represents the rail and B the wheel under test. If this wheel B is not acted on by a brake, it represents a carrying wheel, whereas if subjected to the action of a brake it represents a driving wheel. In the model therefore the conditions of the driving wheel are precisely the reverse of those that occur in practice. But this inversion is of no practical importance because the coefficient of friction does not depend on the direction in which the force is applied. Moreover it makes no difference that we should drive through the rail and brake through the wheel. In order to obtain the conditions of a driving wheel it is only necessary to provide a force acting at the periphery for the trial.

The test wheel B, the dimensions of which are given in figure 3, revolves in a slotted frame C, carried at its upper end by gimbals D, which allow it freedom of movement in the plane of the wheel and also at right angles to this plane. The cord E, passing over the pulley F, subjects the wheel to a force $P_r = P \frac{1\,720}{430} = 4\,P$ pressing it against the disk. A second cord G operated by the screw H serves to apply a lateral force to the slotted frame C. The force P that is applied can be read at any moment on the calibrated spring-balance J. The point of attachment of the cord G is arranged in line with the point of contact of the rolling surfaces.

In determining the loads P the following were taken into consideration in order that the actual wheel-pressures

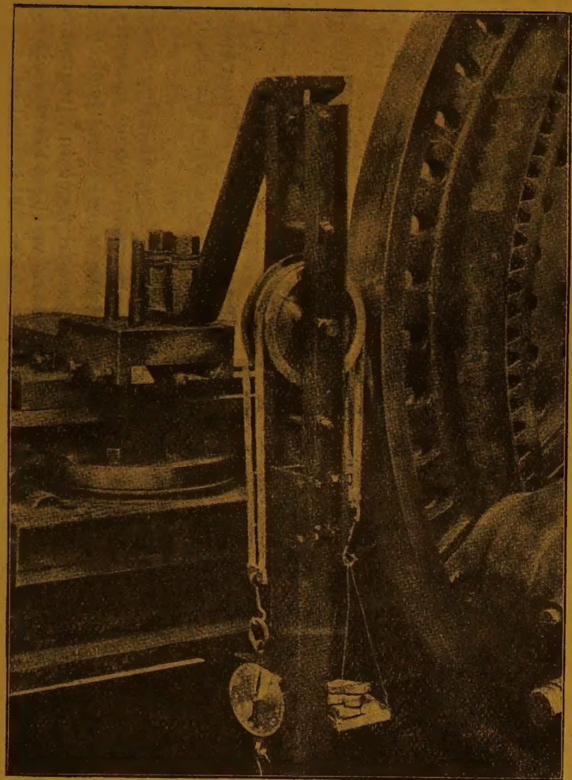


Fig. 1. — View of the trial apparatus.

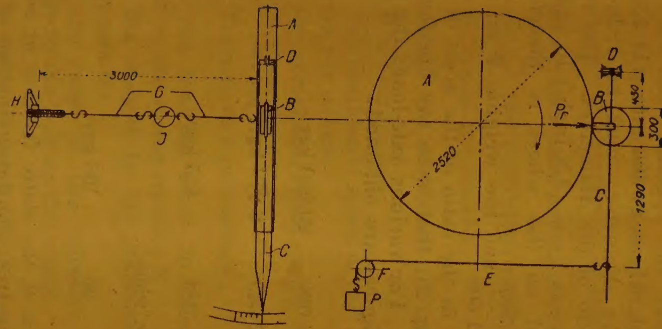


Fig. 2. — Diagram of the trial apparatus.

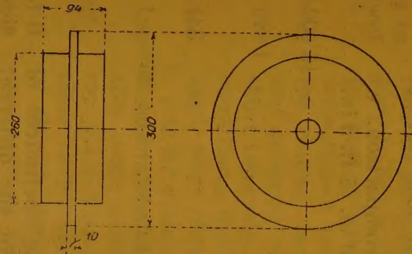


Fig. 3.

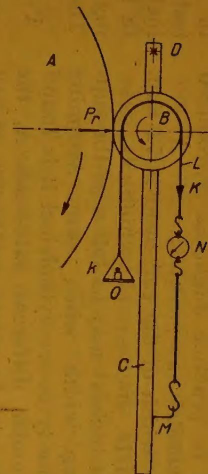


Fig. 4.

Figs. 1 to 4.

found in practice might be reproduced as accurately as possible.

The width of the rail head is 72 mm. (2 13/16 inches) of which 44 mm. (1 3/4 inches) may be taken as the supporting width when new. Assuming that the tyres are cylindrical it is found that under different axle-loads A, the specific loads a per centimetre of rail are $a = \frac{A}{2 \times 4.4}$ kgr. per cm. (lb. per in.). If, as a basis of comparison, we take a driving wheel 1.600 m. (5 ft. 3 in.) in diameter the specific load for the test wheel, which is 300 mm. (11 13/16 inches) in diameter, must be reduced in the ratio of the diameters. The rolling surface is 1 cm. (3/8 inch.) in width. The load on the wheel is therefore $P_r = a \frac{300}{1\,600}$ and the weights P on the cord are in the ratio of the arms of the lever (fig. 2), that is to say one-fourth as great. For various axle-loads the following figures are obtained:

A (1 600 mm. wheel)	=	16 000	18 000	20 000 kgr.
$a = \frac{A}{2 \times 4.4}$.	=	1 820	2 045	2 270 kgr. per cm.
$P_r = a \frac{300}{1\,600}$	=	341	384	425 kgr.
(wheel 300 mm. diameter, rolling surface 1 cm. in width.).				
$P = \frac{P_r}{4}$.	=	85	96	106 kgr.

It was found that the coefficient of friction was the same for the different loads P used during our trial. For this reason, subsequent measurements were only made with smaller weights in order to allow of the use of more accurate spring-balances.

The brake apparatus is shown in figure 4. The test wheel carries a leather band-brake L on both sides of the rolling surface, attached at M to the slotted frame C. If a weight k is placed on the scale-pan O the spring-balance N will give the reading for the tangential force K. The braking force on the diameter of 260 mm. (10 1/4 inches) is therefore (K + the weight on the spring-balance) — (k + the weight of the scale-pan).

The slip was determined by means of a Hasler tangential velocity indicator; with this object the tangential speeds U and u of the disks A and B were measured during the brake tests. The order adopted for the trials was as follows: 1° the determination of the coefficient of friction μ_1 between A and B for $v = 0$, the static friction or adhesion at rest; 2° the determination of the coefficient of friction μ_2 between A and B for $v > 0$, when the brake is not applied to the wheel; 3° the determination of the coefficient of friction $\bar{\mu}_2$ between A and B when $v > 0$, the brake being applied to the wheel.

In the following we shall represent by:

n the number of revolutions per minute of the disk A;

v the tangential velocity of the disk or of the trial wheel in metres per second;

$V = v \times 3.6$ the peripheral speed in km. per hour;

p_1 the lateral force in kilogrammes necessary to move the wheel laterally when $v = 0$;

p_2 the lateral force in kilogrammes necessary to move the wheel laterally when $v > 0$ and the wheel is not braked;

$\bar{p}_2$ the lateral force in kilogrammes necessary to move the wheel laterally when $v > 0$ and the wheel is braked;

P the weight in kilogrammes acting through the cord E;

P_r the load on the wheel in kilogrammes;

$\mu_1 = \frac{p_1}{P_r}$ the coefficient of static friction or the coefficient of adhesion at rest;

$\mu_2 = \frac{p_2}{P_r}$ the coefficient of friction for $v > 0$, (wheel not braked);

$\bar{\mu}_2 = \frac{\bar{p}_2}{P_r}$ the coefficient of friction for $v > 0$, wheel braked;

U the peripheral speed of the disk A in metres per minute;

u the peripheral speed of the test wheel B in metres per minute;

$\frac{u}{U}$ the slip as a percentage;

K the braking force in kilogrammes read on the spring-balance N;

k the weight in kilogrammes on the scale-pan O;

x the ratio of the effective braking effort to the load on the wheel.

The value of the coefficient of friction depends to a great extent on the materials of which the two surfaces in contact are made. For our experiments we have used wrought-iron running on hard grey cast-iron, and for this reason we have obtained somewhat low coefficients of friction. In the trials with which we are dealing, however, we are not concerned with the absolute value of the coefficient of friction, but with its variation at different speeds as compared with its value when $v = 0$. If all the values obtained are referred to the coefficient of static friction the results of the trials may be generalized by expressing the coefficient of friction for any particular speed as a percentage of the coefficient of static friction. Hence the conclusion may be drawn that it is necessary to give particular attention to the determination of the latter. Now the coefficient of friction of any two materials depends, apart from condition of the surface, on the degree of dampness of the air, etc., so that a series of preliminary trials was necessary in order that concordant results might be obtained.

The rolling surface of the disk was turned in the lathe and carefully smoothed with emery. The first values obtained lay so close to each other that they could not be accepted. The explanation was that a series of trials lasted over several days during which the surface underwent change. It was necessary to wash it thoroughly with petrol, to clean it with emery paper, to rub it with brown paper, and then to protect the rolling surface with a clean cloth and proceed with a series of trials at short intervals; it was only under these conditions that figures, that could be used, were obtainable.

For the determination of the static coefficient of friction which was considerably more difficult than that made for $v > 0$ a certain number of readings were taken each time and the mean was taken of these. Moreover, after each individual reading, the wheel was turned and at the same time displaced laterally in order that the line of contact might always be changed. In this way good mean values were obtained.

The results obtained in the trials made with the object of determining the coefficient of lateral friction μ_2 for $v > 0$ and with the wheel not braked (carrying wheel), are shown in table I and are reproduced graphically in figure 5. The trials showed that already at the lowest speed the coefficient of friction μ_2 is sensibly less than μ_1. For $v \geqslant 0$, there is therefore a critical point. At increasing speeds we obtained values which remained constant (83 %) up to V = about 25 km. (15.5 miles) per hour. If the speed were still increased after that point, the coefficient of friction μ_2 at first showed a gradual decrease which became more rapid to fall subsequently to about 70 % at a speed V = 89 km. (55.3 miles) per hour, the greatest speed

Table I. — Determination of the coefficient of friction for $v > 0$: wheel not braked (carrying wheel.)

Trial No. 1.

n Number of revolutions per minute.	V Kilometres per hour.	p_2 Kilo-grammes.	$\frac{p_2}{Pr}$	$\frac{p_2}{Pr}$ as % of $\frac{p_1}{Pr}$
166	79	12	0.1	74
183	87	11.5	0.096	71
187	89	11.2	0.0935	69.5
158	75	11.6	0.0965	71.4
178	84.5	12	0.1	74
168	80	11.5	0.096	71
145	68	12.5	0.104	77
120	57	13	0.1085	80.5
106	50.5	13.5	0.1125	83.5
92	43.8	13.2	0.11	81.5
82	39	13.5	0.1125	83.5
70	33	13	0.1085	80.5
59	28	13	0.1085	80.5
50	23.7	13	0.1085	80.5
39	18.5	14	0.1165	86.5
34	16.2	13.5	0.1125	83.5
30	14.3	13.5	0.1125	83.5
0	0	p_1: 16, 16.5, 16.3, 16 } 16.2	0.135	100

Trial No. 2.

n Number of revolutions per minute.	V Kilometres per hour.	p_2 Kilo-grammes.	$\frac{p_2}{Pr}$	$\frac{p_2}{Pr}$ as % of $\frac{p_1}{Pr}$
112	53	15	0.125	86
124	59	14.8	0.123	85
164	78	12.5	0.104	72
183	87	13.5	0.1125	77.5
160	76	13	0.108	74.5
140	66.5	12.8	0.1065	73.5
130	62	13.2	0.11	76
115	54.5	13.7	0.114	78
100	47.5	13.4	0.1115	77
77	36.5	15	0.125	86
66	31.5	16	0.133	92
60	28.5	15	0.125	86
43	20.4	15	0.125	86
34	16.2	15	0.125	86
25	11.9	13.5	0.1125	77.5
0	0	p_1: 18, 16.5, 17, 18 } 17.4	0.145	100

Trial No. 3.

n Number of revolutions per minute.	V Kilometres per hour.	p_2 Kilo-grammes.	$\frac{p_2}{Pr}$	$\frac{p_2}{Pr}$ as % of $\frac{p_1}{Pr}$
107	51	14	0.1165	78.5
122	58	14	0.1165	78.5
126	60	14.5	0.121	81.5
128	61	14.5	0.121	81.5
168	80	14	0.1165	78.5
174	82.5	13.5	0.1125	75.6
185	88	12.2	0.1015	68.4
188	89.5	12.5	0.104	70
173	82	13	0.108	72.6
154	72	13.5	0.1125	75.6
140	66.5	13.5	0.1125	75.2
124	59	14	0.1165	78.5
110	52	15	0.125	84
90	42.8	15	0.125	84
79	37.5	14	0.1165	78.5
67	31.8	14	0.1165	78.5
60	28.5	15	0.125	84
42	20	15	0.125	84
35	16.5	14.5	0.121	81.5
28	13.3	13.5	0.1165	78.5
0	0	p_1: 17.2, 17.8, 18, 17.5, 18.8 } 17.8	0.1485	100

Trial No. 4.

n Number of revolutions per minute.	V Kilometres per hour.	p_2 Kilo-grammes.	$\frac{p_2}{Pr}$	$\frac{p_2}{Pr}$ as % of $\frac{p_1}{Pr}$
98	46.5	15	0.125	80
122	58	14	0.1165	74.6
132	62.5	13.8	0.115	73.7
138	65.5	12.5	0.104	66.6
162	77	13.5	0.1125	72.2
168	80	14	0.1165	74.6
185	88	12.5	0.104	66.8
187	89	12.5	0.104	66.8
170	81	13	0.108	69.4
152	72	13	0.108	69.4
140	66.5	13	0.108	69.4
122	58	14	0.1165	74.6
108	51	14.5	0.1205	77.3
90	42.7	14	0.1165	74.6
80	38	13	0.1085	69.6
72	34.2	13.2	0.11	70.5
62	29.5	13.5	0.1125	72
50	23.7	14	0.1165	74.6
40	19	14	0.1165	74.6
31	14.7	13	0.1085	69.6
0	0	p_1: 19, 18, 19, 18, 20, 18 } 18.7	0.156	100

possible with the trial apparatus at our disposal. The nature of the curve, however, allows of exterpolation above this speed, so that we can show definitely the coefficient of adhesion for the maximum speed attained in railway working.

When a wheel that is braked is subjected to lateral force there is produced a peripheral force Z (fig. 6) at the point of contact with the rail apart from the lateral force *p*. Before examining the combined results of these two forces, we will first consider the peripheral force Z by itself.

If the tangential force acting on the trial wheel in the direction of K (fig. 4) is arranged to be made just large enough to cause the wheel to slip, the conditions are those of the tangential coefficient of friction at rest $= \mu'_1$ for rotation in the direction of the tangential force. Now the trials have shown that, as might have been anticipated, μ'_1 has the same value as the coefficient of lateral adhesion μ_1. For the wheel at rest we therefore have the ratio $\frac{\mu'_1}{\mu_1} = 1$. If the wheel turns the absolute values of the coefficients of friction change, but the ratio $\frac{\mu'_2}{\mu_2}$ is also unity. Consequently figure 5 gives the values not only of the lateral coefficient of friction μ_2 but also of the tangential coefficient of friction μ'_2 for $v > 0$.

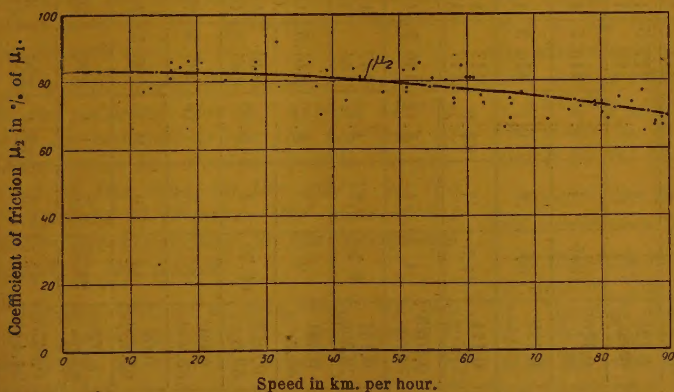


Fig. 5.

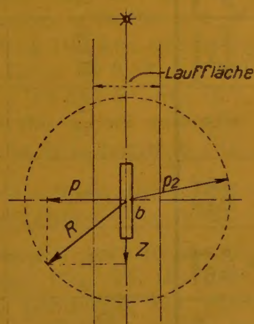


Fig. 6.

Explanation of German term : Lauffläche = Rolling surface.

How are the ratios modified when the two forces *p* and Z act simultaneously ? The trials of the braked wheel throw light on this question. Having already found that the decrease in the coefficients of friction is a function of the speed, it will be sufficient to determine the ratio of the lateral force *p* to Z at a *single speed;* for other speeds it can be calculated directly by means of the curve shown in figure 5.

On account of the heating of the trial wheel we selected a low velocity for the brake trials. At high speeds the temperature of the wheel very rapidly became so high that the moisture that had condensed on the periphery of the wheel evaporated and consequently the surfaces in contact became absolutely dry. The coefficient of adhesion μ_1 was then occasionally doubled in magnitude. After this had been ascertained no further attempts were made to measure braking at high speeds.

The results of trials relating to the coefficient of friction μ_2 for lateral dis-

Table II. — Determination of the coefficient of friction $\overline{\mu}_2$ for $v > 0$; wheel braked, $P = 30$ kgr., $P_r = 120$ kgr.

p_1 Kilogrammes.	$\frac{p_1}{P_r}$	p_2 Kilogrammes.	$\frac{p_2}{P_r}$	$\overline{p_2}$	$\frac{\overline{p_2}}{P_r}$	K Kilogrammes.	k Kilogrammes.	x %	U Metres per minute.	u Metres per minute.	$\frac{u}{U} \cdot 100$	V Kilometres per hour.	K+1.32	K+1.32 − k.	Z Kilogrammes.	$\frac{Z}{P_r}$	$\frac{Z}{P_r}$ as % of $\frac{p_1}{P_r}$.
23 23 25 25 24 23 23 23 21 23 23 22 E. M. 23.2*	0.1935 100 %	20 23 21 21 21 22 18 20 21 17 20 18 E.M. 20.2	0.168 86.7%	17 16 18 17 17 18 18 16 16 16 17 17 E. M. 17	0.1415 73 %	3	1	2.4	117.5	117.5	100	7	4.32	3.32	2.88	0.024	12.4
				E.M. 15*	0.125 64.7 %	6	2	3.85	117.5 117	116 117	100	7	7.32	5.32	4.62	0.0385	19.9
				E.M. 15.1	0.126 65 %	9	3	5.3	117.5 117	117 117	100	7	10.32	7.32	6.35	0.0529	27.4
				E.M. 15.1	0.126 65 %	12.5	4	7.1	116.5	116.5	100	7	13.82	9.82	8.52	0.071	36.7
				E.M. 14.65	0.122 63 %	20	6	11.1	116.5	116.5	100	7	21.32	15.32	13.3	0.111	57.4
				E.M. 13.3	0.111 57.5 %	24.5	7	13.2	117 117	116 116.5	99.5	7	25.82	18.82	15.8	0.1315	68
E. M. 16.5	0.1375 100 %	E.M. 12.56	0.1045 76 %	E.M. 12.3	0.1025 74.6 %	4	1	3.13	146.5	146.5	100	8.8	5.32	4.32	3.75	0.0312	22.7
				E.M. 11.9	0.101 73.5 %	6.5	2	4.2	147 146	146.5 148	100	8.8	7.82	5.82	5.02	0.0422	30.7
				E.M. 10.9	0.091 66.2 %	9	3	5.22	146.5	146.5	100	8.8	10.32	7.32	6.27	0.0523	38
				E.M. 10.5	0.0875 63.6 %	11.75	4	6.55	146 145.5	146.5 146.5	99.5	8.8	13.02	9.02	7.84	0.0653	47.5
				E.M. 10.8	0.09 65.4 %	13.5	5	7.13	295.5	295	99.8	17.75	14.82	9.82	8.54	0.0714	52
				E.M. 10.8	0.09 65.4 %	17	6	9.35	295.5	295	99.8	17.75	18.32	12.32	10.7	0.081	59
				E. M. 9	0.075 54.5 %	21.5	7	11.4	296	295	99.6	17.75	22.82	15.82	13.7	0.0975	71
				E.M. 5.67	0.0473 34.4 %	28	9	15.4	296	Limit of friction.	0	17.75	29.32	20.32	18.5	0.154	112

(*) E. M. = mean figure; in the first line we reproduce all the experimental figures: in the following lines, to save space, we only show the means of a whole series of figures obtained experimentally.

placement with $v = 0$ and with the braked wheel (driving wheel condition) are shown in table II, and are reproduced graphically in figure 7. As in the case of μ_2 in figure 5, the results are expressed as percentage of $\mu_1 = \mu'_1 = 100\ \%$. It will be seen from figure 7 that the coefficient of friction diminishes when the tangential force Z increases. How is this decrease to be explained ?

The wheel is subjected to the action of two forces, the tangential force Z and the lateral force p (fig. 6) which are at right angles to each other and act at the point of contact b. If the resultant of the two forces (1) p and Z, is taken as R (fig. 6) this latter is approximately constant and also agrees with lateral force p_2 for the wheel that is not braked. The result of the investigation consequently gives rise to the following important conclusion :

If a rolling wheel is subjected to forces at its point of contact, a state of slip will be reached when the resultant of these forces has a value equal to the product of the load on the wheel by the coefficient of friction; the value of the latter corresponding to the peripheral speed of the wheel.

Having arrived at this result we can now proceed to state the conditions. If we draw an arc of a circle (fig. 8) of radius 100 % $= \mu_1 = \mu'_1$, having the zero value at the centre, we shall thus have defined the resultant corresponding to the conditions of rest for the wheel. Figure 5 shows that from the commencement of the smallest rolling movement the coefficient of friction falls to about 83 %. We shall therefore draw, as in the previous case, an arc of a circle of 83 % radius and we shall thus obtain the resultant for V = 0 to 20 km. (0 to 12.4 miles) per hour. The radii of these circles represent higher velocities which diminish proportionately to the diminution of μ_2 (fig. 5).

The abscissæ and the ordinates nevertheless do not always represent coefficients of friction, but they may also be regarded as forces if a given value $\mu_1 = 100\ \%$ is taken. Suppose for example that $\mu_1 = \frac{1}{3} P_r$, which is the case for a good dry condition of the rail, then for a pair of driving wheels with a load of 18 tons one obtains with p and Z = 100 % a value of 6 tons (13 230 lb.). At the speed of 100 km. (62.1 miles) per hour the maximum tractive effort of an electric locomotive having four driving axles, for example, is 7 000 kgr. (15 432 lb.) or 1 750 kgr. (3 858 lb.) per axle, that is 29.2 % of 6 metric tons. If in figure 8, we draw a vertical at the abscissa 29.2 %, its intersection with the arc of the circle representing 100 km. per hour will give a lateral force of $0.59 \times 6\,000 = 3\,540$ kgr. ($0.59 \times 13\,230 = 7\,805$ lb.), necessary to move the wheel laterally, that is to say to cause it to begin to slip.

Apart from the main trials, the following subsidiary observations were made :

If the disk is caused to turn in the direction of the arrow (fig. 2), the case of a hauled axle is produced, but with the pivoting point in front. Left to itself the trial wheel remains in the neutral position. By the neutral position we mean that for which the supporting arm C (fig. 2) is exactly vertical and the axis of the trial wheel parallel to the axis of the disk A.

To obtain the *smallest* deviation it was necessary to apply the forces shown in table I. When the force had caused the wheel to deviate to a certain extent and was then suddenly removed, the wheel returned instantaneously after a fraction of a revolution to its neutral position.

This fact is therefore a contradiction to the theory that it is necesary that the axle that is running should be subjected to a movement of deviation before it

(1) This is to say the values obtained simultaneously during the trials.

offers resistance to friction. *To cause lateral movement of a hauled wheel from its neutral position a force of definite magnitude is necessary, the value of which is equal to the product of the load on the wheel multiplied by the coefficient of friction corresponding to the actual peripheral speed of the wheel.*

By causing the disk to turn in the opposite direction to the arrow (fig. 2) the case is that of an axle that is pushed. Left to itself the wheel, however accurately it was set, did not remain in the neutral position, but always wandered to the right or to the left. *It follows definitely that the hauled wheel is in a condition of stable equilibrium and the pushed wheel in a condition of unstable equilibrium.*

It is possible that the results of these trials may not be accepted without criticism. It may be objected that a trial made on a model does not allow conclusions to be drawn that are directly applicable to practice. To this we should reply that an accurate solution of the problem that presents itself could never be obtained on an actual locomotive when running or could only be obtained at the expense of an enormous amount of time and money; and that, moreover, the methods employed for carrying out these trials were so chosen that the results could easily be compared with those of practice.

The objections, for example, that a railway axle carries two wheels whereas

Figs. 7 to 10.

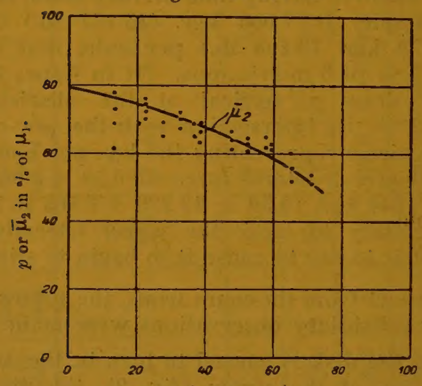


Fig. 7.

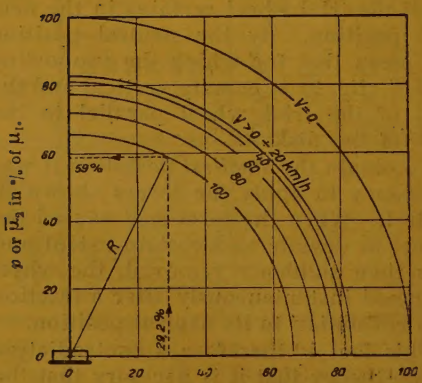


Fig. 8.

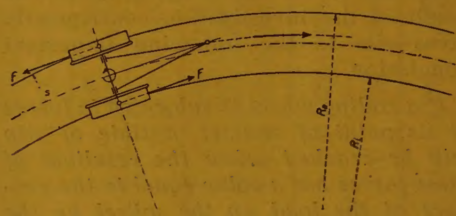


Fig. 9.

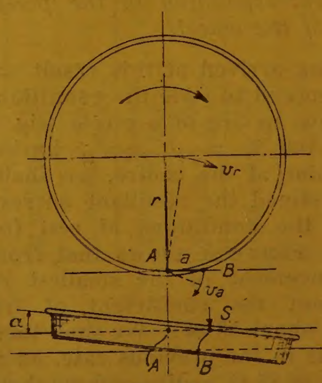


Fig. 10.

the axle in the model only carries one wheel, or that the conical form of the tires has not been copied in the model are not sufficient to deprive the results of the trials of their value, because a pair of wheels forms a cylinder of which the central portion is wanting; the test wheel used in the experiments must similarly be considered as a cylinder. With regard to the conical form of the tires this only exists when they are new. After a comparatively short period of time the surface of the tire has lost its original form. Consequently the question of conical form cannot be considered as an absolutely essential factor in determining the movement of the wheel on the rail.

It might also be objected, that we have not taken account in our trials of the influence exerted by a pair of wheels of equal diameter running over a curve; the friction then produces a couple $F \times s$ (fig. 9) which tends to cause the axle to turn in a horizontal plane. This rotational couple is caused by the advance of the inner wheel over the outer wheel. The tires should therefore, neglecting the elastic flexibility of the axles, of the spokes of the wheels and of the rail, be in a continuous state of slipping and simultaneously of rolling. This continuous condition does not exist in practice, but the wheel has an alternating movement of rolling and then for a short period of slipping, because the axle acts the part of a torsion spring which is twisted by the couple $F \times s$ and twists back when the force F becomes so great as to reach the limit of adhesion. This state of slipping and rolling of a pair of wheels on the rails has been the subject of lengthy study and investigation by Mr. A. Wichert in the periodical *Verkehrstechnik* of March 1921. Actually the influence of this rotational couple can be neglected, particularly in the case of the matter before us, the main object of which is to throw light on those problems concerning the running conditions that have the greatest bearing in practice; that is to say in our opinion, those that relate to normal-gauge locomotives for express trains. The radii of the curves with which one has to deal in this case vary from 300 m. to 1 000 m. (15 to 50 chains). The most unfavourable conditions occur with a curve of radius of 300 m. on the assumption that the tires are cylindrical and of equal diameter. Calculation shows that the *displacement by slipping represents one-half per cent of the displacement by rolling.* On the other hand the ratio of the time of slipping to the time of rolling is appreciably smaller, so small in fact that the effect of slipping disappears before that of rolling. The atmospheric conditions, dry weather, rain, ice, and fog produce important changes in the coefficient of friction. This variation is shown with remarkable accurracy in the trials on the model; in fact, by raising the temperature of the wheel B to about 60° C. (140° F.) it was possible nearly to double the coefficient of friction as compared with normal conditions (10° C. to 15° C. [50° F. to 59° F.]). This also affords an explanation of the fact that the application of the brakes for some time may appreciably improve the adhesion of the driving wheels in the case of difficulty in starting.

The results of our trials which may also be applied to automobiles are given, we repeat, not as absolute figures, but as relative figures, so that by the judicious selection of the class of friction under investigation, all the conditions that occur in practice may be suitably reproduced.

Application of the data obtained.

By means of the data obtained we shall now investigate the behaviour of the various arrangements of locomotive-axles with regard to the pressure on the flanges; of the effect of this pressure on the wheel, and on the rail, and on the actual rolling movement.

When running on a curve the wheels of each vehicle tend to bear against one line of rail, or to arrange the axles radially, according to whether the plane of the wheel makes an angle with the radius of the curve which is unequal to 90°, or equal to 90°. The method of calculation usually adopted up to the present for ascertaining the pressure on the flanges and the corresponding wear is incorrect. We give below a more accurate calculation based upon the results we have obtained.

Each pair of wheels tends to advance in the direction of the plane of the wheel. If the latter makes an angle other than 90° with the radius of the curve the flange touches the rail at a definite angle α, preventing the wheel from proceeding in the direction it has and compelling it to following the curve of the rail; the wheel slips in the direction of the radius of the curve and pressure is produced on the flange amounting to $P_r \times \mu_2$. *This figure is independent of the angle of incidence α, it depends only on the load on the axle and on the coefficient of friction corresponding to the particular rolling speed.*

If a wheel proceeds in the direction of the arrow (fig. 10) it turns about an instantaneous centre A. The pressure S of the flange at B, gives rise to a braking force $S = \mu_g$ acting against the rolling movement, where μ_g represents the coefficient of friction against sliding The instantaneous velocity v_a of the point B is $\frac{V_r a}{r}$ the loss of power W_r due to S, will therefore be $W_r = \frac{\mu_g \times \mu_2 \times P_r \times v_a}{75}$ metric H P.; or, if we replace v_a by $\frac{V_r a}{r}$ and we express the speed in kilometres per hour, instead of giving it in metres per second:

$$W_r = \frac{\mu_g \times \mu_2 \times P_r \times V_r \times a}{270 \cdot r}.$$

The unknown quantity a is the horizontal distance at which the lateral force between the flange and the rail acts, measured from the vertical axis through the centre of the wheel, and can be determined at any distance. By means of wooden full-size models it has been measured with great exactitude for normal rail and tire sections by taking impressions of the surfaces in contact; figures 11 and 12 show this for angles of incidence α of 1° and 2°, and for wheel diameters of 950 mm. and 1.600 m. (3 ft. 1 3/8 in. and 5 ft. 3 in.).

The losses due to friction at various speeds have been taken in figure 13 for these conditions on the assumption that the load on the small axle amounts to 12 tons and that on the large axle to 18 tons. The coefficient μ_g of sliding friction is that given in *Hutte* (vol. I, p. 243*b*, 3rd example). The character of these curves is of interest because they show in the first instance that an increase of losses occurs up to a speed of about 70 km. (43.5 miles) per hour for both wheels, above this speed the curve drops again. The reason for this is that the lateral coefficient of friction μ_2 and particularly the coefficient of sliding friction μ_g diminishes with increased speed. It will be noticed that *the angle of incidence does not affect the pressure of the flange.* On the other hand *the frictional losses* for a wheel that runs along a rail *vary with the magnitude of the angle of incidence α.* The frictional losses for the axles which take up an oblique position are therefore actually very large and may, particularly in the case of wheels of large diameter and heavily loaded, lead to abnormal wear of the flanges and necessitate premature renewal of the pair of tyres. It is obvious that this wear on the tires has its counterpart in the wear of the rail heads and gives rise to heavy work on maintenance of the track. *In the case of leading wheels, and particularly of driving wheels having a large angle of incidence, the use of a good system for greas-*

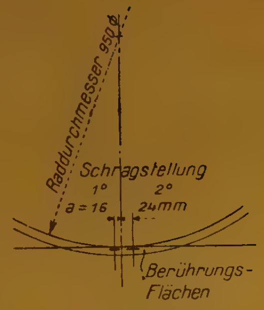


Fig. 11.

Fig. 12.

Explanation of German terms of the figures 11 to 14 : Raddurchmesser = Diameter of wheel. — Schragstellung = Oblique position. — Berührungsflächen = Contact surfaces. — Triebrad = Driving wheel. — Laufrad = Carrying wheel. — Neuer Spurkranz = New flange. — Abgenutzter Spurkranz = Worn flange.

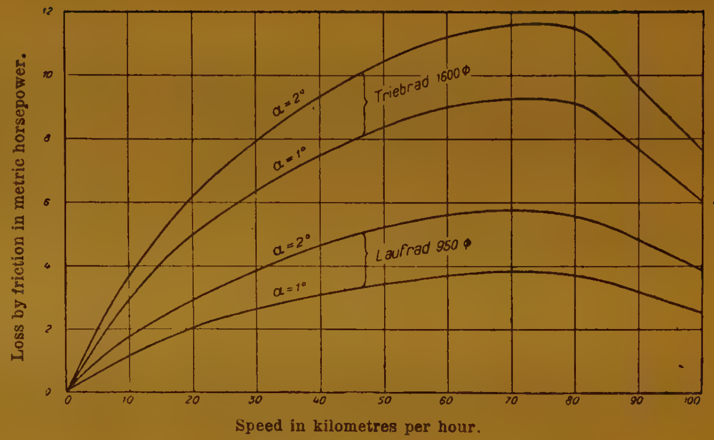


Fig. 13.

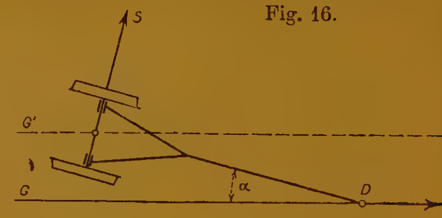


Fig. 16.

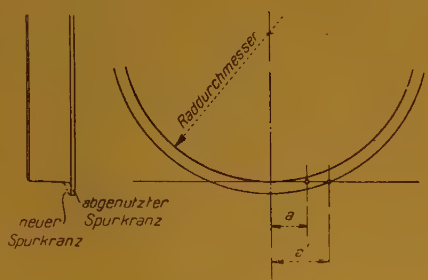


Fig. 14.

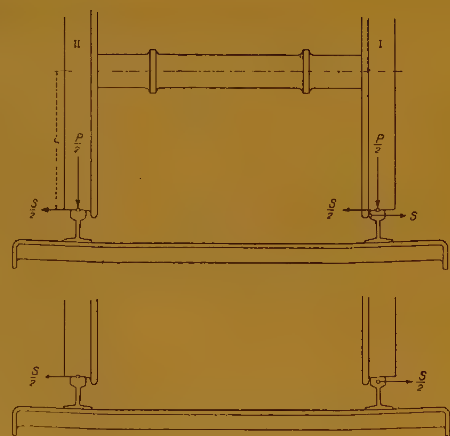


Fig. 15.

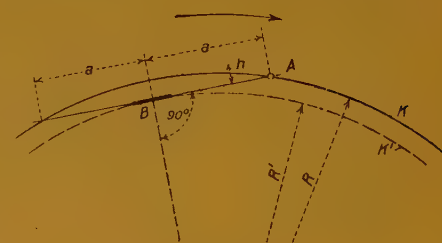


Fig. 17.

Figs. 11 to 17.

ing the flanges is to be strongly recommended.

Wheels having a large angle of incidence have also other disadvantages. For the same angle of incidence, a (fig. 14) is greater with worn tires; the result of this is not only to increase the loss due to friction, but at the same time to increase the tendency of the flange to climb on to the head of the rail, particularly at the rail-joints and point-tongues. From the preceding it follows that the axle arranged radially can carry a much greater pressure on the flange than one that is arranged obliquely to the radius. *From the point of view of the pressure on the flanges and of smooth running on the curves, the radial axle has great advantages over an axle that takes a position inclined to the radius.*

We will also briefly examine the influence that a pair of wheels rolling in an oblique direction exercices on the track. The flange that rubs or presses sideways against the head of the rail with a force S (fig. 15) the magnitude of which can be calculated exactly. The force S is the reaction necessary to overcome the resistances to friction $\frac{S}{2}$ produced on the wheels in consequence of the obliquity of the axle.

It is understood of course that the forces S and $\frac{S}{2}$ acting on the head of the outer rail must be of opposite sign; consequently the force $\frac{S}{2}$ with which the wheel II presses outwards against the left rail head remains.

Axles occupying an oblique position when rolling cause forces to act on the rail head due to friction and these forces tend to increase the gauge of the track and cause forces of reaction to act on the axis of the pair of wheels tending to cause bending and having a couple equal to $\frac{S}{2} \cdot r$. The stress resulting from this is of course of considerable magnitude; it is, however, generally neglected in calculations for the strength of axles.

Making use of the data that have been obtained up to the present we will now investigate how the different arrangements of axles on locomotives work from various points of view; that of their rolling properties, that of the pressure on the flanges, and that of the action of these pressures on the wheel and on the track. A knowledge of the behaviour of the articulated frame with one and two axles will suffice to enable this to be understood also in the case of trucks having more than two axles. The arrangement of the wheels on any vehicle is, actually, nothing more than an arrangement of carrying trucks or motors having one or more axles.

The best known radial axles are those of Bissel and Adams. They are identical and several other types are also identical with them so far as their arrangements for rolling and obliquity of the axis of the axle is concerned. The axle is connected to the main frame in such manner that it can deviate sideways with regard to the latter by turning about a pivot arranged either to the front or to the rear and on the longitudinal axis of the locomotive. If the pivot is placed in front of a trailing axle when running forward the axle will occupy a *radial position*. The path followed by the pivot definitely determines that to be taken by the axle. If, for example, the pivot D moves along the straight line G (fig. 16) the pair of wheels can only be drawn along a parallel line G′ under the action of a force S, acting towards the outside, and equal to the product of the load on the axle by the coefficient of friction. When G′ coincides with G and consequently $\alpha = 0$ the force S tending to bring it into line is no longer necessary. If the pivot in advance of the axle describes an arc of a circle of radius R (fig. 17) the centre B of the axle describes an arc of a circle

of radius $R' = R - h$ where h is the sagitta of the arc and where $R' = \sqrt{R^2 - a^2}$, when a represents the length of the sagitta of the Bissel bogie (fig. 16).

If the direction of running is reversed (fig. 16) the force S will change in sign and it will be necessary for it to act towards the inside of the curve in order that the pair of wheels should continue to follow the straight line G′ parallel to the direction of running. When G′ coincides with G the force S tending to keep the wheels in line should vanish because the axle when in this position is in unstable equilibrium. The slightest deviation under this condition would momentarily cause the axle to return to stable conditions provided that the friction of the flange against the rail did not prevent it. These facts show at once in their practical application that a leading Bissel truck when running on the straight must have a great tendency to lateral oscillation in consequence of the play that exists between the flange and the rail. The check-springs would not prevent this additional movement unless the force exerted by them attained the magnitude of $P_r \cdot \mu_2$: in some circumstances, on the other hand, they might damp out the sideway oscillation of the whole locomotive. On a curve, according to the build of the locomotive and the length a of the deflection of the bogie, a leading Bissel truck will rub its flanges against the outer or the inner rail (figs. 18 & 19).

Fig. 18.

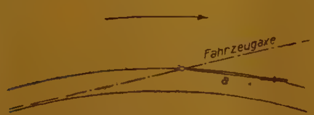


Fig. 19.

Explanation of German term :
Fahrzeugaxe = Centre line of vehicle.

In the case of locomotives required to run on a line having numerous curves the designer should so choose the axis of rotation that the leading axle will definitely exert pressure against the outer rail. This arrangement will assure the avoidance of sideway oscillation, at any rate on curves.

The truck with *two axles*, or the bogie, is distinctly superior to the Bissel bogie in respect to its directional properties. Contrary to the case of the single axle truck, the position of which between the tracks is determined by the driving axles to the rear of it, it exercises a determining influence on the running. Actually it has no tendency to produce sideway oscillation either on the curve or on the straight. On the straight the wheels of the two axles, owing to any accidental cause, may rub against the one or the other line of rail and remain in this position until change of level of the track or accidental forces acting at the pivot drive them back towards the other rail. The oblique position cannot be maintained and consequently *periodic* sideway oscillations (swaying) cannot occur.

We have seen from figure 17 that the path of a trailing axle B is definitely determined by that of its central point of attachment A. The function of this point is fulfilled in the ordinary bogie by the leading axle. On a curve it follows the outer rail with an angle of incidence that varies with the wheel-base of the bogie and the radius of the curve dependent on the clearance. If the line follows a circle of radius R the axle B describes a circle of radius R′ determined by the expression $\sqrt{R^2 - a^2}$ where a represents the wheelbase.

Figure 20 shows in what manner the trailing axle II follows the leading axle I, carried on the same frame, when the latter suffers to and fro sideway movements, within the limits of play between the flange and the rail due to bad state of the track or other disturbing influences. The zigzag line 1 represents, for

example, the movement of the leading axle I, the curve 2 shown in dot and dash gives the path taken by a point D on the bogie between the two wheels I and II : and the curve 3 is the path of a point situated at D′ at the centre of the wheel II. From this it may be deduced that *in the case of short and sudden sideway deviations of the leading axle, such as occur in service, each point of*

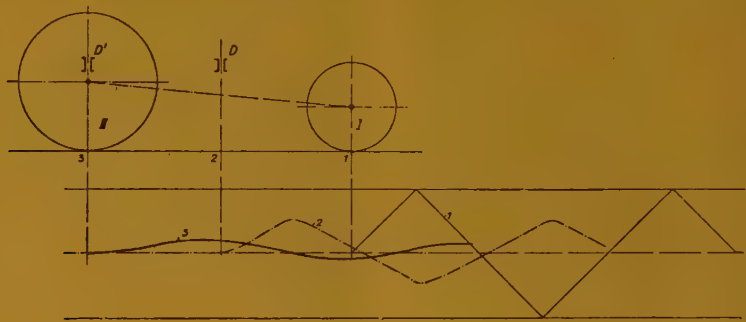


Fig. 20.

the bogie is subjected to a similar deviation the magnitude of which is less the nearer the point is to the trailing axle.

The sideway oscillations of the front axle which extend over a longer period of time and over a longer distance also involve a much greater sideway displacement of the trailing axle II, but this displacement can never be produced of such magnitude that it is necessary to make provision against rebound of the flange from the rail. *Consequently, the second axle of a bogie runs much more smoothly and with much more stability than the leading axle over any lateral irregularity in the track.*

The determination of th best position for the pivot of the American four-wheel bogie used almost exclusively on express locomotives and on eight-wheel passenger coaches is still to-day a controversal question. It will be seen from figure 20 that steadier running and less wear of the flanges and of the rails will occur when the pivot is placed above the trailing axle II, the loads on the axles being so distributed that the sum of the centrifugal forces and of the accelerating forces acting on the pivot when taking a curve remain definitely less than the friction $P_r \cdot \mu_2$. The ideal arrangement of the axles of a 40-ton coach, for example, would be that shown in figure 21. This method of construction is particularly suitable for the direction of running shown in this figure, but as vehicles of this type cannot be turned end for end the symmetrical construction with central pivots is preferred. In the case of steam locomotives this objection does not apply consequently there is no objection to the use of the unsymmetrical bogie.

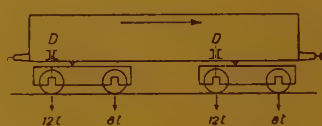


Fig. 21.

There is still a word to be said regarding the *bogie brake.* Opinions are greatly divided with regard to its use. Many administrations do not fit it; others have adopted it for many years. No report has appeared dealing with its advantages and disadvantages that has not raised some objection to its use. In view of the continued increase in the speed of trains and of the forces developed when elec-

tric motors are used for high speeds, the necessity for using all the wheels of a locomotive for braking becomes imperative.

The application of the brakes to a pair of wheels reduces the lateral adhesion of these on the rails. Figure 7 shows the magnitude of this decrease when the braking forces are increased.

It has been proposed that the pressure on the brake blocks of express electric locomotives should be increased so that the ratio of the brake pressure on the wheel to the load on the wheel would appreciably exceed unity. As a result of this the force S tending to keep a pair of wheels in place on the track is almost neutralized. *A locomotive of which the brakes are applied hard on all the wheels loses its lateral reaction and floats between the limits of play allowed by the track.* On the straight the effect of braking does not produce any serious difficulty; on the other hand, the stability of running is appreciably less on a line that has a number of reverse curves run over at high speed. Running off a curve to the right on to a curve to the left, or entering and leaving a curve, are always accompanied by violent shocks against that rail which produces the change of direction; greater wear of this rail results and, in some cases, may involve risk of derailment. *It is therefore advisable that the leading bogies of locomotives hauling express trains should only be braked to a moderate extent.*

There are also bogies having *more than four wheels* of which the axles are parallel and are carried in the same frame. The middle pair of wheels may with advantage have a certain amount of lateral play relatively to the side frames. This is done with the object of avoiding abnormally high pressures on the wheel flanges, of increasing the amount of adhesion of the vehicle on the rails, and of obtaining better running on curves.

We will consider, for example, a six-wheel vehicle (fig. 22). On a curve the axle I runs along the outer rail with an angle of incidence α and as we know it brings into action a correcting force S equal to the pressure against the rail.

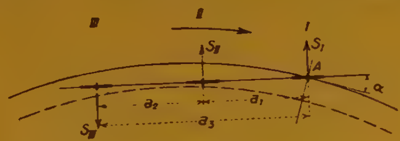


Fig. 22.

Axles II and III have each a tendency to take up a radial position; but, in consequence of their parallel arrangement on the common bogie frame, they cannot simultaneously assume a radial position and the result is to bring into action correcting forces S_{II} and S_{III} at the ends of the lever arms a_1 and a_2 which tend to cause the chassis to turn about A the point of contact of axle I. S_{II} and S_{III} have the same value and are of opposite sign. Consequently the correcting force acting on axle III is diminished by the quantity $S_{II} \cdot \frac{a_1}{a_3}$. If, on the other hand, axle II can move laterally with regard to the chassis it will rub against the outer rail and exercise no influence on the running of axle III; in other words the vehicle will have the rolling properties of a four-wheel bogie of which the wheelbase will be a_3. The same arguments apply equally to bogies with eight and ten wheels and as each locomotive represents a combination of several trucks their mutual action contributes to determining the smoothness of the running with regard to the wear of the track and stability when running.

The use of an independent drive of the axles of electric locomotives enables considerable subdivision to be made of each frame. Use is often made of this feature to improve the running on curves; sometimes the designer omits to take sufficient account of the influence of this arrangement and in particular the

type 2-4+4-2 (fig. 23) has been suggested as giving the best results in express train working. The connexion between the two motor-bogies 1 and 2 may be so chosen that the independent *lateral* freedom of movement is completely preserved;

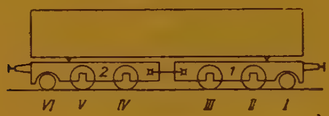


Fig. 23.

but it often happens that connexions are used which prevent this freedom of sideplay. The former arrangement, with the heavy loads carried on electric locomotive wheels, gives excessive pressure on the flanges and excessive wear of the wheels on axle IV. The consequence of the latter arrangement is a less smooth running of the vehicle because at high speeds the locomotive has a considerable tendency to acquire a sideway oscillating movement.

The explanation of this is given in figure 24. This shows that the forces tending to keep the axles III and IV radial are equal and opposite because III and IV are connected at the point M in such manner that the sideway deviation of the two bogies, that would naturally occur, is prevented; these two axles, are in a position of unstable equilibrium with regard to transverse displacement within the limits of play *t* in the track, and this instability would be disturbed by the slightest change in the coefficient of friction or by other accidental causes. The two axles « float » as it were from side to side within the limits of play *t* because the sum of the forces exerted in keeping them in radial position has become equal to zero.

The reader will be enabled to understand the methods of calculating the righting forces and the pressures on the flanges, as found from the results of the trials, by a numerical example with which we shall conclude. This deals with an electric locomotive for express trains with independently driven axles of the type 2-4-0+0-4-2. The four motors are arranged on the main frame. The loads on the axles are shown in figure 25. The two end axles are combined with the carrying axles in a four-wheeled bogie; the pivot (fig. 26) is placed immediately behind the driving axle; the two motor axles in the centre have some lateral play.

In order to show more clearly the difference between this new arrangement and the ordinary types we shall proceed to make a comparison with a locomotive in which the axles follow in the same order, but which is fitted with similar bogies of the well-known Krauss-Helmholtz type (fig. 27). The use of the word bogie for the Krauss-Helmhotz method of construction is not strictly accurate the two axles form parts of different frames. It is rather a question here of a special form of Bissel bogie (fig. 28) the running of which on the straight differs in no essential from that of the ordinary Bissel, whereas there is an improvement in running on curves and particularly in entering the curve due to the play of the driving axle and the displacement of the pivot. An advantage of this method of construction over that shown in figure 27 is that the properties mentioned determine the automatic sub-division of the centrifugal forces applied to the pivot D between the two axles. Moreover, it is to be noted that the driving axle of the Krauss-Helmholtz bogie, owing to the large angle of incidence that it makes on curves and of the large amount of frictional work which this causes, would be incapable by itself of absorbing the whole of the centrifugal force. But there need be no hesitation about transferring this load to the driving axle of the truck which is shown in figure 26, particularly when it runs as the leading bogie. It is, moreover, always possible to divide the effect of centrifugal force as desired be-

tween the two axles by means of the check springs.

In figures 29 and 30 relating to the locomotive shown in figure 25 we have shown, diagrammatically, the pressures of the tyres that are produced when running round a curve of radius R = 1 000 m. (50 chains) at a speed of 105 km. (65.2 miles) per hour. The vehicle shown in figure 29 has bogies with the pivot carried towards the rear (fig. 26) and that in figure 30 has carrying bogies of the Krauss-Helmholtz type (fig. 27). We have not taken account of wind pressure or of the other lateral forces which may accidentally be brought into play because it is necessary to make arbitrary assumptions in order to determine them, and because under normal conditions their effect is small in comparison with that of the centrifugal forces.

We have selected a curve of 1 000 m. (50 chains) radius because at the higher limit of speed legally permitted this is the radius on which the effective centrifugal forces attain their maximum. In the case under consideration this effective centrifugal force amounts to 4 200 kgr. (9 260 lb) and represents the portion of the total centrifugal force to be taken into account in the determination of the pressures on the flanges. Allowance by deduction for the corresponding super-elevation has consequently been made.

The manner in which the locomotive fitted with the bogies shown in figure 26 conforms to the curve can be seen in the following way : the outer circle represents a curve of radius R = 1 000 m. On the inside 10 mm. (3/8 inch) has been allowed for total play between the flange and the head of the rail. The wheels of axle I rub; axle II is arranged radially. In case of radial position it is found by calculation that the distance between the outer rail and the flange is 1 mm. (3/64 inch). From these data the position of the front bogie can be determined. With regard to the rear bogie, the wheels on axle V touch the outer rail and axle VI is convergent, so that the position of this bogie can also be determined. If we now join the two centres of rotation D_1 and D_2 whe shall obtain the longitudinal axis of the locomotive. It will be found that the distance of the centre of axle I from the centre of the locomotive is 12 mm., and 16 mm. (15/32 and 5/8 inch) relatively to axle VI. Axles III and IV should have a lateral play of 15 mm. (19/32 inch) on each side of their normal positions. Consequently each pair of wheels runs independently without exercising any force on the side frames. The wheels on axle III rub against the heads of the outer rails, those on axle IV against the head of the inner rail. Centrifugal force is absorbed partially by means of the check springs acting on axles I and VI which are capable of withstanding a load of 1 000 kgr. (2 205 lb.) each, so that 1 200 kgr. (2 645 lb.) is carried on each of the two pivots.

As axle I makes an angle with the rail, an effect tending to bring it radial is produced to which, as the result of the measurements we have made, we can assign a numerical value. We see from figure 5 that for V = 100 km. (62.1 miles) the coefficient of friction μ_2 diminishes to 66 % of the value of μ_1. Supposing that $\mu_1 = \frac{1}{3}$ we find that when the carrying axle carries a load of 12 tons the force tending to bring the axles radial is $\frac{1}{3} \times 12 \times 0.66 = 2\,640$ kgr. (5 820 lb.). To this force due to the righting forces must be added the load on the springs, that is to say 1 000 kgr. (2 205 lb.), hence a total pressure on the flanges of 3 640 kgr. (8 025 lb.) is obtained. Axle II is arranged radially and the pressure on the flanges is therefore nil. This axle however undergoes the action of centrifugal force amounting to 1 200 kgr. at the centre of rotation D_1. Is adhesion sufficient to resist this force amounting to 1 200 kgr.? Here again we are taking

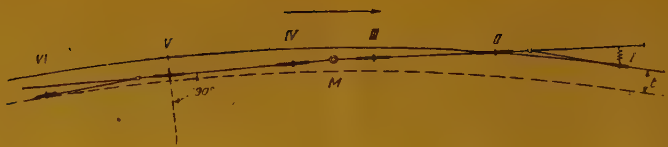


Fig. 24. — Position taken by the locomotive of figure 23 on a curve.

Fig. 25.

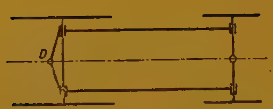


Fig. 26.

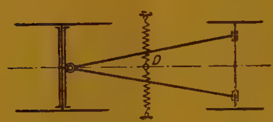


Fig. 27.

Fig. 28.

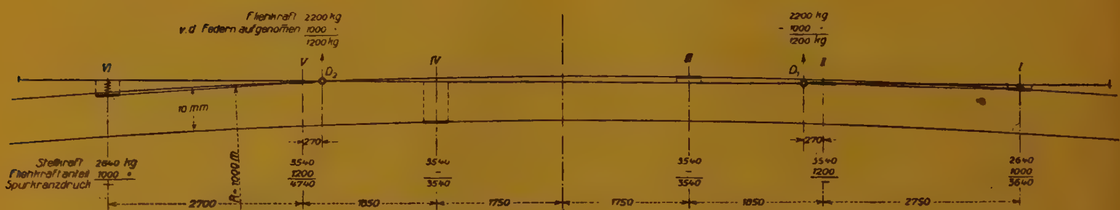


Fig. 29. — Locomotive having the axle arrangement of figure 25 and the bogies of figure 26 in position on a curve.

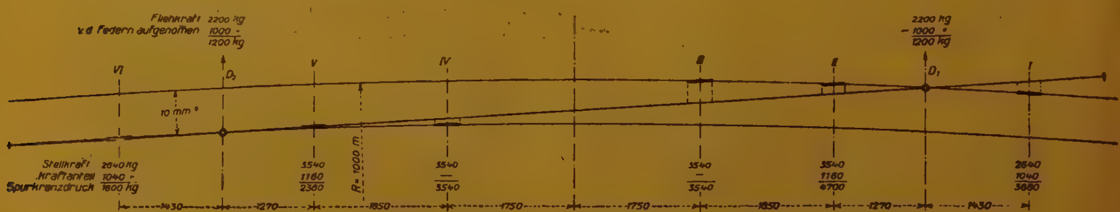


Fig. 30. — Locomotive having the axle arrangement of figure 25 and the bogies of figure 27 in position on a curve.

Explanation of German terms of the figures 29 and 30 : Fliehkraft = Centrifugal force. — v. d. Federn aufgenommen = Absorbed by the check springs. — Stellkraft = Righting effort. — Fliehkraftanteil = Share of the centrifugal force. — Spurkranzdruck = Pressure exerted by the flange.

the most favourable case. Let us assume that the locomotive is working at full load, developing a maximum tractive effort of 7 000 kgr. (15 432 lb.) or 1 750 kgr. (3 858 lb.) per axle. The total tractive effort at the drawbar is supposed to be equal to one-third of the adhesive load that is to say 6 000 kgr. (13 230 lb.) of which 1 750 kgr. represents 29.2 %. Consequently, as shown by figure 7 when V = 100 km. per hour, for an effort at the periphery of the wheel of 29.2 % there is a reduction in the righting force of 59 %. This force is therefore $\frac{1}{3} \times 18\,000 \times 0.59 = 3\,540$ kgr. (7 805 lb.). The righting force being approximately 2.5 times as great as the maximum centrifugal force there is no cause for sideway oscillation. Axles III and IV follow obliquely to the radius of the outer and the inner rails. The pressure of the flanges is 3 540 kgr. in both cases. (See axle II). Axle V also runs in an oblique position. The righting force amounts, as in the other cases to 3 540 kgr. With the 1 200 kgr. which must be added to it for centrifugal force the pressure on the flanges is raised to 4 740 kgr. (10 450 lb.). Axle VI runs radially, the pressure on the flanges is therefore equal to zero. The righting force exceeds the centrifugal force by 1 640 kgr. (3 615 lb.).

The same locomotive fitted with bogies of the type shown in figure 27, gives, other things being equal, the arrangement on the curve shown in figure 30. We shall commence, as we did with figure 29, by fixing the positions of the bogies which then determine that of the locomotive. The centrifugal force acts at the axes of rotation and is distributed in proportion to the lengths of the arms of the lever between the carrying axle and the driving axle of the bogie.

Axle I arranges itself obliquely and is subject to the same righting force as above, that is 2 640 kgr. (5 820 lb.). To this must be added the portion of the centrifugal force representing 1 040 kgr. (2 293 lb.) which gives a total pressure on the flange of 3 680 kgr. (8 113 lb.). Axle II also runs obliquely, hence there is a righting force of 3 540 kgr. (7 805 lb.); taking into account the share of the centrifugal force the total pressure on the flange is obtained and amounts to 4 700 kgr. (10 360 lb.). Axles III and IV behave as in figure 29. The wheels on axle V rub against the inner rails, the share of the centrifugal force not being sufficient to overcome the righting force. The pressure of the flange is 3 540 kgr. (7 805 lb.) less the share of the centrifugal force amounting to 1 160 kgr. (2 557 lb.) that is 2 380 kgr. (5 248 lb.). The same is the case with axle VI for which the pressure of the flange is accordingly 2 640 — 1 040 = 1 600 kgr. (5 820 lb. — 2 293 lb. = 3 527 lb.).

It follows from these considerations that locomotives fitted with bogies of the types shown in figure 26 would doubtless have advantages from the points of view of stability in running and of maintenance over locomotives with carrying bogies constructed on the general lines of the type shown in figure 27.

[621 .133.3]

Locomotive boiler performance,

By E. C. POULTNEY, O. B. E.

Figs. 1 to 15, pp. 442 to 455.

(*The Engineer.*)

The coal consumption of a locomotive per unit of power developed for any given amount of steam supplied to the cylinders largely depends on the proportions of the boiler and the quality of the fuel. The amount of heating surface and the size of the grate determine the evaporative power of the boiler, and upon the proportions which exist between the heating surface and the area of the fire-grate depend other characteristics of importance. The results obtained by a change in the relationship between the amount of heating surface for each square foot of grate are important, and it ~~further~~ constitutes one of the chief variations ~~which~~ are found when the dimensions of boilers of different locomotives are examined.

With the object of bringing to notice the performance of locomotive boilers differing in the proportions of their related parts when operating at various rates of power output, the following article is offered. The information given is based upon tests made by the Pennsylvania System on the locomotive testing plant at Altoona, Pa., and is taken from Bulletins Nos. 18 and 21, dealing respectively with tests made on « Pacific » and « Atlantic » type express locomotives, classes « K 2 s a » and « E 6 s, » Nos. 877 and 89. The writer is indebted to Mr. J. T. Wallis, chief of motive power, for his courtesy in allowing this information to be published. The tests reports give complete information as to the action of the locomotives, but in this article, only the performance of the boilers and superheaters will be considered. Further, it is not intended to go into the subject in its entirety; that is, it is not proposed to deal with all details relative to the expenditure of the heat units in the fuel fired, and conclude with a complete heat balance, but rather to discuss the more important actions which go to make up the boiler performance. After a description of the locomotives involved, the following different functions of the boilers and superheaters will be considered, each under a separate sub-heading, which are designated as follows :

1° Blast action and draught;

2° Sparks;

3° Fire-box and smoke-box temperatures;

4° Evaporation and fuel consumption;

5° Steam pressures;

6° Boiler efficiencies;

7° Heating surface action and heating surface grate area ratios.

8° Long and short boiler tubes;

9° Superheater performance;

10° Fuel consumption and power developed.

General description of the locomotives.

The following is a general description of the two locomotives, the particulars given relating more particularly to the boilers :

« *Pacific* » 4-6-2 *Type.* — The « Pacific » type locomotives, class « K 2 s a », were built at the Juniata shops, Altoona, Pa., and were the first superheated 4-6-2 type engines to be constructed by the Pennsylvania system. They are fitted with boilers having a Belpaire type fire-box, with a straight topped barrel section and a circular smoke-box 79 1/2 inches long inside. A steam dome is mounted on the rear barrel course. The fire-box has sloping back and throat plates, and is fitted with a brick arch carried on four arch tubes. The grate has an area of 53.72 square feet, of which about 25.71 square feet is the active shaking area. Drop grate sections, 12 inches wide, are placed at both front and back ends, and along the centre line is a centre grate bearer, having air inlets, and which is 7 3/4 inches wide on the top face. The barrel is fitted with a Schmidt type superheater of thirty-two elements, having a diameter of 1 7/16 inches outside, and a heating surface (fire-side) of 989.32 square feet. The flue tubes are 5 1/2 inches and 2 1/4 inches outside diameter, and the length over tube sheets is 21 feet (252 inches). The smoke-box is equipped with a superheater compartment, with an automatic damper. The exhaust nozzle is rectangular, 4 3/4 inches by 7 inches, the top of which is 12 3/4 inches below the centre line of the boiler. A cast iron lift pipe extends from the centre line to the chimney. No change was made in the front end arrangement during the progress of the tests.

« *Atlantic* » 4-4-2 *Type.* — The « Atlantic » type engines, represented by class « E 6 s, » were the first engines of that type to be built having superheaters, and followed a class known as « E 6 » working with saturated steam. Both were alike, with the exception of the superheating equipment. The boiler has a Belpaire type fire-box and a combustion chamber of medium length, and has sloping back and throat plates. The barrel is composed of two courses, a parallel section next the fire-box and a tapered section next the smoke-box. The largest diameter is 83 1/2 inches outside. The length of the smoke-box is 69 1/2 inches inside and between tube sheets 164 5/8 inches (13 ft. 8 5/8 in.). There are 242 2 inch tubes and 36 5 3/8 inch flues, and a Schmidt type superheater is fitted, having 1 1/2 inch tubes. A steam dome is situated on the rear barrel section. The fire-box is fitted with a brick arch carried on three water tubes. The fire-grate is generally similar to that fitted to the K 2 s a « Pacific » engine, and has an area of 55.23 square feet.

The ratio of proportions — as set out in the table in the next column — show how the two boilers differ in the size and disposition of their heating surfaces. The « Pacific » type engine, owing to the fact that it has three pairs of wheels under the barrel portion of the boiler, has considerably more tube surface than the « Atlantic » type, having the shorter barrel section. The tubes for the « Pacific » engine are 250.08 inches long; those for the « Atlantic » engine being only 164.63 inches, their outside diameter being 2.25 inches and 2.0 inches respectively.

The total heating surface (fire-side) in the tubes, flues and superheaters of the two boilers is 4 104.38 and 2 835.01 square feet for the « Pacific » and « Atlantic » type engines, so that the « Pacific » type engine has 44 % more heating surface in the tubes, flues and superheater than the « Atlantic » locomotive. The « Atlantic » engine has the larger fire-box of the two. This is due to the fact that it is fitted with a combustion chamber,

which, together with the three arch tubes, gives it a total heating surface of 254.48 square feet, against 208 square feet for the « Pacific » type engine, which is not fitted with a combustion chamber, but which has four arch tubes.

Table of general dimensions and proportions.

Heating surfaces :	« Pacific » type, class « K 2 s a. »	« Atlantic » type, class « E 6 s. »
Tubes and flues (fire-side)	3 115.06	2 146.20
Fire-box and arch tubes .	208.02	254.43
Superheater (fire-side). .	989.32	688.81
Total, combined. . .	4 312.40	3 089.49
Grate area	53.72	55.23
Fire-box volume, cubic feet.	244.3	256.18
Heating surface based on water-side of tubes and flues	3 436.37	3 348.19
Tubes :		
Number	202	242
Outside diameter, inches .	2.25	2
Length, inches	250.08	164.63
Flues :		
Outside diameter, inches .	5.5	5.375
Length, inches	250.08	164.63
Cylinders	24 × 26	22 × 26
Drivers, diameter, inches .	80	80
Boiler steam pressure. . . .	205	205

Boiler proportions, heating surface based on fire-side tubes and flues.

Total heating surface / Grate area . . .	80.2	55.8
Superheater surface / Total heating surface	22.9	22.3
Fire-box heating surface / Grate area . .	3.88	4.6
Fire-box heating surface / Total heating surface . .	4.83	8.24
Fire area through tubes and flues / Grate area	12 %	14 %

Boiler proportions, heating surface based on water-side of tubes and flues.

Total heating surface / Grate area . . .	86.2	60.6
Superheater surface / Total heating surface . . .	21.3	20.5
Fire-box heating surface / Grate area . .	3.88	4.6
Fire box heating surface / Total heating surface . .	4.5	7.6
Fire area through tubes and flues / Grate area	12 %	14 %

Coal used.

The coal used throughout the tests for the two locomotives was Penngas, a bituminous fuel having the following characteristics :

« Pacific » type engine tests : Average approximate analysis.

Fixed carbon, %.	56.77
Volatile matter, %	35.08
Moisture, %.	1.09
Ash, %	7.06
	100.00
British thermal units (dry), per lb. .	14 530

« Atlantic » type engine tests : Average approximate analysis.

Fixed carbon, %.	58.45
Volatile matter, %	33.65
Moisture, %.	1.54
Ash, %	6.36
	100.00
British thermal units (dry), per lb . .	14 470

The calorific value of the coal fired was for all the tests very uniform. For the « Pacific » type locomotive the variation was from a maximum of 14 735 British thermal units to a minimum of 14 207 British thermal units per lb., dry, and in the case of the « Atlantic » the

figures were from 14 698 British thermal units to 14 156 British thermal units per lb.

Blast action and draught.

The « Pacific » type engine tested had a chimney 18 1/2 inch diameter, increasing to 19 inches at the top, and the exhaust nozzle was rectangular, 4 3/4 inches by 7 inches, the longer dimension being lengthwise of the locomotive. An investigation of the exhaust action showed that with this form of exhaust nozzle the chimney was completely filled during rates of actual evaporation from 18 000 lb. to 47 000 lb. of water per hour. At the latter rate a draught of 15 inches of water was attained in the smoke-box at the front of the diaphragm. The dynamic pressure indicated across the top of the chimney being at the same time 4.5 lb. per square inch.

Tests of a locomotive No. 150, class « K 2, » exactly similar to the « K 2 s a » class, but without superheater, and having in consequence considerably more water heating surface, showed that with a circular nozzle, 6 1/2 inch diameter, a greater evaporation than 44 900 lb. could not be obtained with a maximum draught of 8.5 inches and a dynamic pressure across the chimney of 7 lb. This pressure was not uniform, rising to a peak at the centre of the chimney, showing that it was not properly filled. On a rectangular, 7 inches by 4 3/4 inches nozzle being applied, the dynamic pressures obtained were more uniform, and at 4 lb. per square inch and with a draught of 13.9 inches the evaporation was increased 19 % to 53 300 lb.

The « Atlantic » type locomotive, when first the tests were undertaken, was fitted with a circular exhaust nozzle, 6 1/4 inch diameter. With this it was found not possible to obtain a higher rate of evaporation than 36 600 lb. of water per hour. At an actual evaporation of 32 700 lb. there was a high peak of pressure of about 32 lb. per square inch across the top of the chimney, diminishing to about 5 lb. round the sides. The smoke-box draught was 8.5 inches. To improve the chimney conditions a rectangular nozzle of equal area to the circular nozzle was fitted, with the result that the dynamic pressures were more equalised, and the possible evaporation was increased 7 % to 38 800 lb., with a smoke-box draught of 12.9 inches. The chimney had a diameter of 17 inches, increasing to 19 inches at the top. No definite reason is advanced to account for the better draught condition obtainable with a rectangular, as against a circular nozzle, but it is suggested that they direct the exhaust steam and entrained gases, so that they more completely fill the chimney and cause a larger volume of gas to be drawn out.

Figures 1 and 2 show the draught records for each boiler respectively, the draught in inches of water being plotted against the rate of firing (dry coal) per square foot of grate per hour. The four curves drawn indicate the draught readings at the ashpan, fire-box and behind the diaphragm — that is, at the smoke-box tube plate — and also in the smoke-box in front of the diaphragm. Within limits, an increase in draught is followed by an increase in the combustion rate, and the results shown in the case of the « Atlantic » seem to indicate that more fuel might have been fired to advantage; reference, however, to figure 5 giving the rate of evaporation shows, however, that above a firing rate of about 7 000 lb. of fuel per hour, further coal was not burned, as the rate of evaporation did not increase.

The space between the lines indicates the resistance between the different sections of the two boilers, and the «Pacific» type boiler having the long tubes shows the greatest draught loss to take place between the fire-box and the smoke-box tube plate. The draught readings in the

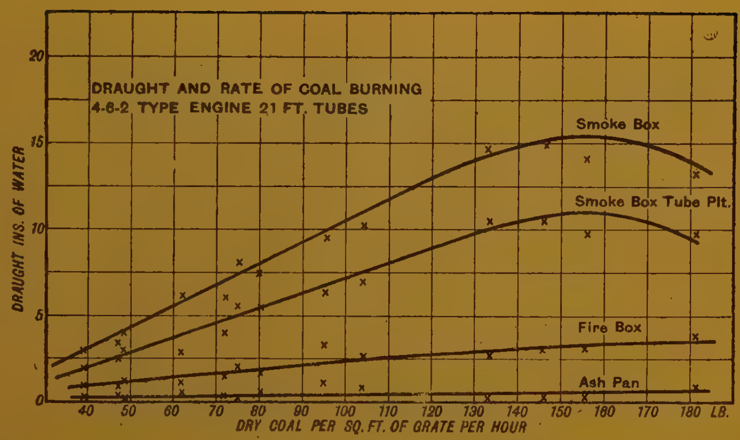


Fig. 1.

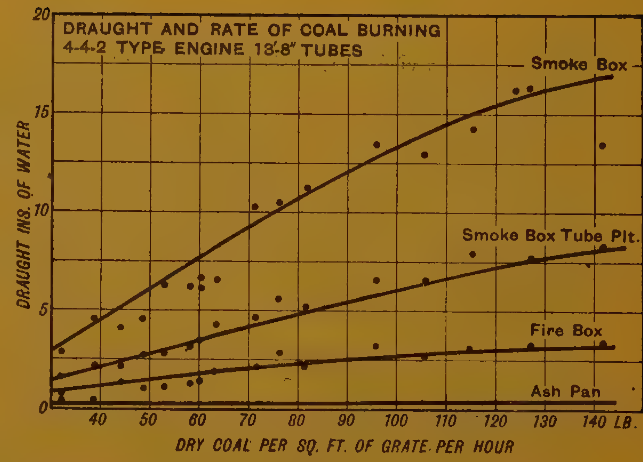


Fig. 2.

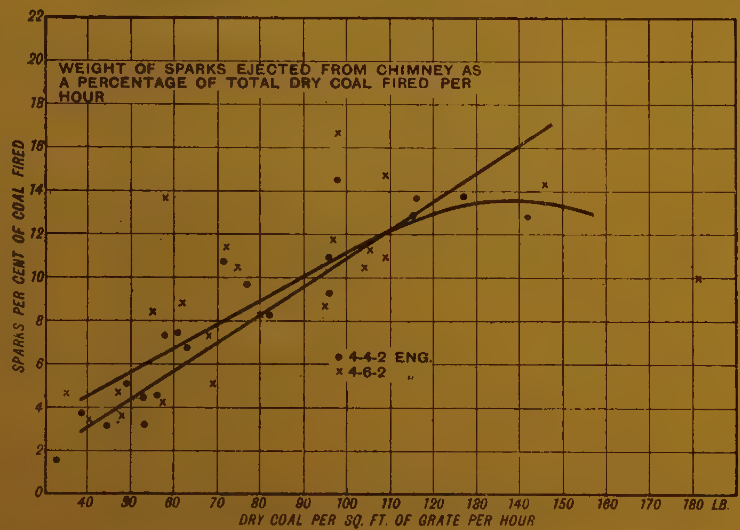


Fig. 3.

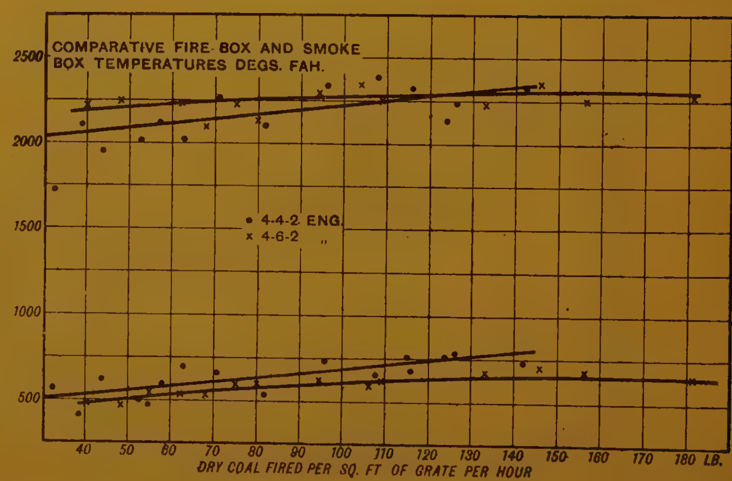


Fig. 4.

ashpans are in each case low, indicating adequate air inlets.

For the « Pacific » the ashpan air inlet was 10.8 square feet or 21 % of the grate area, and for the « Atlantic » 8.3 square feet, equal to 15 % of the grate area.

The draught readings taken in the fire-box indicate the resistance caused by an increase in the rate of firing owing to the thicker fire carried. The value of the other readings is also seen to increase, owing to the larger quantity of gas to be moved with the greater weight of fuel fired. When, however, more coal is fired than can be burned — as is shown by figure 1 — then the draught falls. This, as will be mentioned is due to the decrease in the evaporation which naturally follows the failure of combustion.

The difference between the draught readings for the « Atlantic » type boiler between the back and front of the diaphragm is attributed to the gas passages being inadequate.

The « Pacific » type boiler was forced to higher rates of combustion than the « Atlantic, » the greater quantity of steam generated, because of the more extensive heating surfaces, enabling an increase in draught to be obtained. At the highest rate of firing of 181 lb. of dry coal per hour, the equivalent evaporation was about 57 000 lb. of water per hour, against 51 000 lb. for the « Atlantic » at 142 lb. per square foot of grate per hour.

Spark losses.

One of the greatest heat losses in the action of the locomotive boiler is that due to the weight of the sparks discharged. The tests of the « Atlantic » locomotive were the first made at Altoona, when it was possible by means of a special spark catcher erected on the top of the test plant to collect everything discharged from the chimney except the very fine dust which escaped with the steam and smoke. As the firing rate increased, it was established that there was a rapid rise in the weight of sparks ejected from the chimney.

For the « Atlantic » engine, the sparks discharged amounted to as much as 1 000 lb. at a firing rate of 7 000 lb. of dry coal per hour; whilst for the « Pacific » the weight of sparks reached 1 466 lb. when 7 183 lb. of dry coal were fired.

In figure 3 an attempt has been made to plot the weight of sparks ejected from the chimney against the fuel fired per square foot of grate per hour. The weight of sparks being expressed as a percentage of the total coal fired per hour, the percentage lost for the « Atlantic » engine seems to increase in a uniform manner as the rate of firing increases, and a straight line can apparently be drawn which fairly represents the rate of increase.

For the « Pacific » the points are much more widely scattered, and it seems that the spark losses diminish at the higher rates of firing. A curve can hardly be drawn to represent the results fairly, so the line shown scarcely does more than indicate what would appear to be the general result for this locomotive. Analysis of the sparks obtained during the trials of the « Atlantic » locomotive indicated that their value per pound in British thermal units ranged from 9 800 to 12 097.

Fire-box and smoke-box temperatures.

The temperature in the two fire-boxes ranged from 2 000° to 2 400° Fah., at firing rates of 30 lb. to 140 lb. of dry coal per hour, corresponding to 1 670 lb. to 7 900 lb. of coal per hour for the « Atlantic » engine, and for the « Pacific » from 2 200° to 2 300° Fah. at firing rates of from 30 lb. to 180 lb. of coal per square foot of grate per hour, corresponding to from 1 600 lb. to nearly 10 000 lb. of coal per hour.

The comparative fire-box and smoke-box temperatures are given in figure 4,

the curves indicating that at all rates of firing the smoke-box temperature is lower in the case of the « Pacific » type engine.

Another point of interest brought out by the curves is the fact that at the higher rates of combustion those above 100 lb. of coal per square foot of grate per hour, the fire-box and smoke-box temperatures increase only slightly, and become almost constant as the firing rate further increases, and, as will later be shown, this is also true of the temperature of the superheated steam.

The heat absorbed by the heating surfaces of the two boilers was about 1 550° for the « Atlantic » type, and about 1 700° for the « Pacific » type; and the differences between the fire-box and smoke-box temperatures being uniform for all rates of firing suggests that the boiler heating surfaces absorb a fixed amount of heat from each cubic foot or from each pound of the gases flowing over them, and that therefore increase in evaporation is due to increase in the weight or quantity of these gases and not to their temperature.

Steam pressures.

With few exceptions, throughout all the tests both boilers maintained the working steam pressure or 205 lb. per square inch. For the « Pacific » locomotive the highest average pressure was 206.6 lb. and the lowest 196.4 lb.; at the latter figure the boiler was evaporating its maximum weight of water of 64 711 lb. (equivalent evaporation) per hour, and the rate of combustion was 146.7 lb. of dry coal per square foot of grate per hour. The speed was (240 revolutions) 55 miles per hour, the regulator was wide open, and the cut-off 50 %, the power developed being 2 411 indicated horse-power. The « Atlantic » maintained a highest average pressure of 205.9 lb. and a lowest of 183.1 lb. At the highest equivalent evaporation attained of 52 084 lb. of water per hour, the firing rate was 115.5 lb. of dry coal; the maintained steam pressure was 205.5 lb. In this case the speed was (360 revolutions) 84 miles per hour, cut-off 25 %, and the regulator was full open. Under these conditions the power obtained was 2 350 indicated horse-power.

Test No. 2839 was made at the same speed, 84 miles per hour, but at a cut-off of 30 %, and it is interesting to note how an attempt to increase the power output brought about a reduction in the maintained steam pressure, which in this instance was 192.5 lb. The equivalent evaporation obtained was 51 241 lb. per hour, with a firing rate of 142 lb. of dry coal per square foot of grate, and the power obtained was 2 355 indicated horse-power. The coal rate was increased 24 %, but the equivalent evaporation only increased 0.016 %.

The change in the cut-off caused a stronger draught at the smoke-box tube plate, as would be expected, the draught obtained being 8.2 instead of 7.9, but the greater rate of firing resulted in a reduction in the equivalent evaporation per pound of fuel fired from 8.16 lb. to 6.53 lb., and the boiler efficiency fell from 55.9 to 44.7 %. The increase in the power output was only 5 indicated horse-power. This was obtained, as stated, by lengthening the cut-off, which, it should be noted, raised the steam rate from 16.0 lb. to 16.27 lb. per indicated horse-power hour.

Evaporation and fuel consumption.

Figure 5 shows the equivalent evaporation per square foot of combined heating surface (fire-side of tubes and flues) at various rates of firing (dry coal) per square foot of grate per hour.

It will be noticed that the evaporation power of each square foot of heating surface in the case of the « Atlantic » type boiler is much in excess of that obtained with the « Pacific » type. The « Atlan-

tic » reaches its highest rate of evaporation when 115 lb. of dry coal are fired per square foot of grate, when the equivalent evaporation reaches 16.8 lb. per hour per square foot of heating surface in the boiler and superheater. Higher rates of firing do not produce greater rates of evaporation, the additional coal fired is evidently not burned. The « Pacific » type boiler can be made to evaporate 15.1 lb. of water per square foot of heating surface per hour when the fuel rate is 146 lb. of dry coal per square foot of grate per hour. At a firing rate 181 lb., or nearly 10 000 lb. of dry coal per hour, the evaporation is only about 13.5 lb. of water.

The evaporative performance per square foot of heating surface is therefore not so good as in the case of the « Atlantic » type boiler. In this connection it must be remembered that the ratio of grate area to total heating surface must be considered. For the « Pacific » this figure is 80.2, whilst for the « Atlantic » it is but 55.2. The « Atlantic » evaporates more water per square foot of heating surface, and therefore for its size is the more powerful boiler; but the « Pacific » type boiler, having the larger heating surfaces proportional to the grate area, is for any given rate of combustion the more economical boiler, and its superiority in this respect increases as the rate of firing becomes greater.

For the « Pacific » type boiler the maximum rate of firing required a total of 9 764 lb. of dry coal to be fired per hour, whilst for the « Atlantic » 7 800 lb. was the maximum weight used. The boilers were fired at this rate for 15 minutes. At the higher rates of firing they had both to be operated with their fire-doors open. If they were closed there was always an immediate drop in steam pressure, showing a diminishing rate of evaporation. To obtain the best results with this method of working, it was found desirable to run with a green fire at the fire-door in order that the flame might heat the air as it was drawn through the doorway. If this method was not followed, the steam pressure could not be kept up. Not enough air could be drawn through the grate when the boilers were forced to the highest rates of output. The difficulty seems to be that air supplied cannot be distributed through the burning coal properly. The particulars of the air supply passages for each boiler are shown in the following table.

Table of air inlet areas, square feet.

	« Atlantic » type.	« Pacific » type.
Through fire-box sides . .	0.017	0.02
— grate	15.19	16.34
— fire-door	0.06	0.07
— ashpan	10.8	8.3
— grate area . . .	53.72	55.23
$\frac{15.19 \times 100}{53.72}$ = ...	29.6 %	...
$\frac{16.34 \times 100}{55.23}$ = ...	...	29.6 %
$\frac{10.8 \times 100}{53.72}$ = ...	21 %	...
$\frac{8.3 \times 100}{55.23}$ = ...	...	15 %

In the very noticeable falling off in evaporation shown by the curve for the « Pacific » type boiler, it seems probable that an explanation can be found for the decrease in the sparks discharged from the chimney at the higher rates of firing. It will be noticed that the highest evaporation reached was 15.1 lb., equal to a total equivalent evaporation of about 64 000 lb. per hour. When working at this rate the draught at the smoke-box tube plate was — from figure 1 — about 15 inches of water. An increase in the rate of firing failed to produce a corresponding increase in evaporation ,and when 181 lb. of dry coal was fired the equivalent evaporation obtained was only about 13.7 lb.

per square foot of heating surface, or about 57 000 lb. per hour. The smaller quantity of steam discharged through the exhaust lowered the smoke-box draught to 13.3 inches, and, the action on the fire being diminished, had apparently the effect of reducing the weight of sparks thrown from the chimney. Figure 6 shows how the equivalent weight of water evaporated per pound of dry coal fired per square foot of grate per hour varies with the equivalent evaporation per square foot of heating surface. Here the « Atlantic, » having the smaller heating surfaces relative to the grate area, has the advantage from the point of view of economical steaming, though, as the rate of evaporation increases, the water rate per pound of coal fired falls away more rapidly than is the case with the « Pacific » type boiler, which, as already noted, can be forced to a higher rate of coal consumption. The advantage shown by the « Atlantic » type locomotive is due, of course, to the fact that for any given rate of evaporation per square foot of heating surface the coal rate per square foot of grate area per hour is lower.

Referring back to figure 5, it will be noticed, for example, that for an equivalent evaporation of, say, 13 lb. of water per square foot of heating surface, the « Atlantic » type boiler burns but 76 lb. against about 109 lb. per square foot of grate per hour for the « Pacific » type, corresponding to 1.360 lb. and 1.355 lb. per square foot of heating surface respectively. For a given rate of combustion the boiler having the larger heating surfaces relative to the grate surface has been shown to be the more economical; but on the basis of a given evaporation per square foot of heating surface, the boiler having the larger grate in proportion to the heating surface is to be preferred.

The two additional lines drawn in figure 6 show how the rate of evaporation and the equivalent water per pound of dry coal compare with the results obtained with two locomotives of the same type and similar to K 2 s a and E 6 s, but working with saturated steam. K 2 is a « Pacific » type engine having a boiler with a total heating surface of 4 156.91 (fire-side of tubes) and 53.3 square feet of grate, against a total heating surface, evaporative and superheating, of 4 312.4 square feet for K 2 s a, and E6 has a total heating surface 3 197.02 square feet (fire-side of tubes) and 54.75 square feet of grate, against 3 089.49 square feet of heating surface, boiler and superheater, for E 6 s. The saturated steam locomotive K2 has slightly less heating surface than K 2 s a fitted with superheater, and the « Atlantic » E 6 has 108.5 more than E 6 s.

The equivalent evaporative efficiency of the boilers without superheaters is greater than those of the same general dimensions, and having appoximately equal heating surfaces, evaporative and superheating combined. The addition of a superheater of the Schmidt or flue tube type necessitates a reduction in the water heating surface, and, further, the elements absorb part of the heat which would otherwise be transferred across the evaporative heating surfaces. This, and the fact that the superheating surfaces are not so efficient in the absorption of heat as the waterheating surface, account for the lower efficiency of the boilers fitted with superheating equipment of the type mentioned.

Boiler efficiencies.

The comparative boiler efficiencies per cent are shown by figure 7. The « Pacific » type boiler is more efficient than the « Atlantic » type for all rates of firing, and the efficiency of the latter falls off more rapidly as the rate of combustion increases. Further, for any given efficiency the larger boiler can be forced to a higher rate of combustion.

Additional lines, indicating the effi-

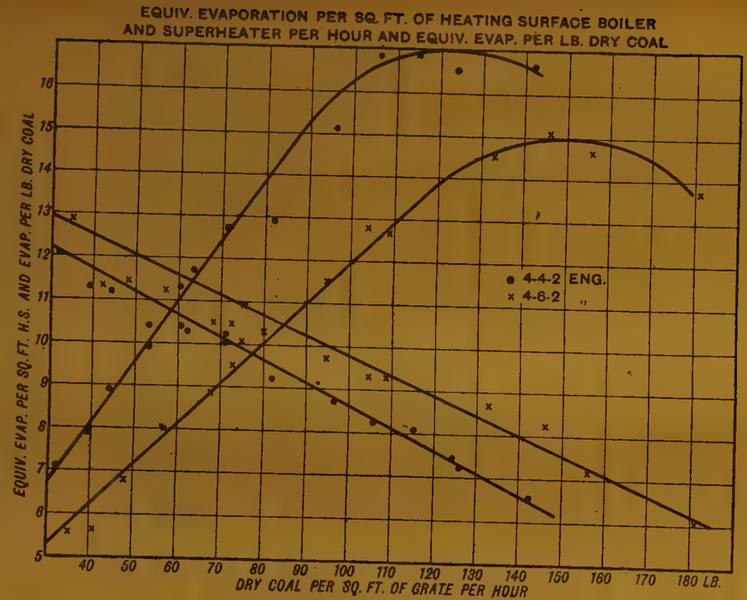


Fig. 5.

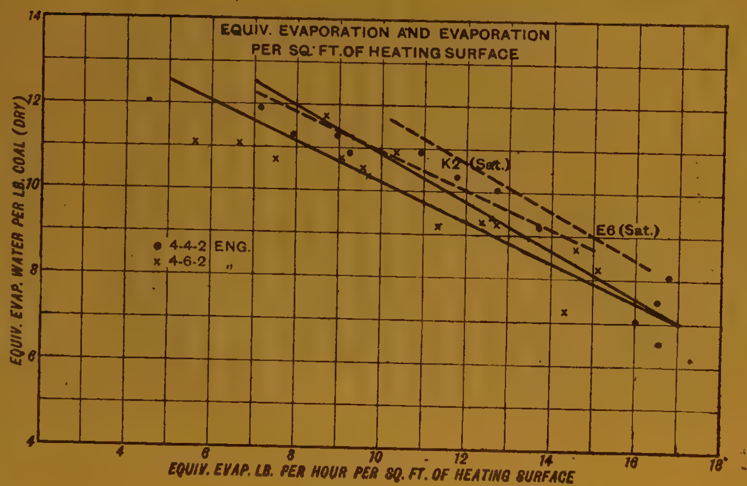


Fig. 6.

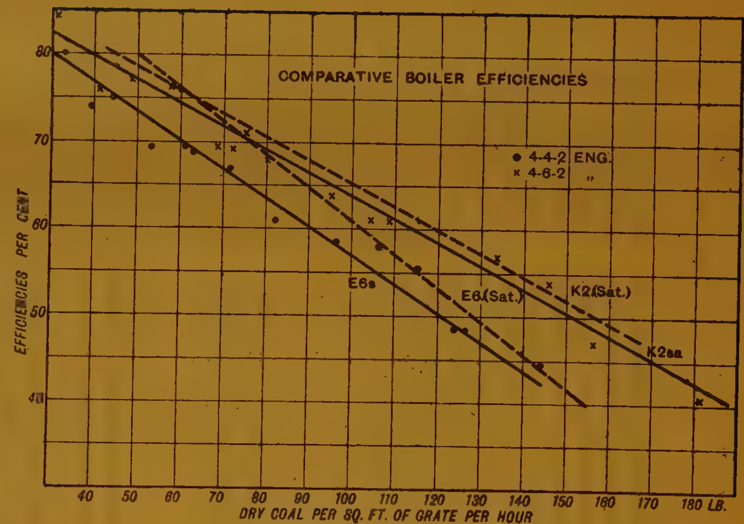


Fig. 7.

ciency of the two boilers, K 2 and E 6, not fitted with superheaters, have been included. Their thermal efficiencies are higher than those shown for the superheater-fitted boilers of the same general size. While it is true that the performance of the boilers supplying saturated steam are the most efficient, it is also true that any losses in the performance of the superheater-equipped boilers is more than made up by the greatly improved engine efficiencies realised.

***

We will now discuss the action of the evaporative heating surfaces in absorbing the heat liberated by the burning fuel on the grate.

Action of evaporative heating surfaces and heating surface-grate area ratios.

From the preceding sections, especially those having reference to the fire-box and smoke-box temperatures and the comparative evaporations obtained, the following are the conclusions arrived at regarding the action of the evaporative heating surfaces in absorbing the heat available.

On the basis of a given rate of combustion for the boiler having the larger heating surfaces, a less quantity of fuel is fired per square foot of heating surface, which means that a less quantity of gas is formed relative to the heating surfaces; therefore each cubic foot or each pound of gas passes over a larger amount of heating surface, each square foot of which absorbs a given amount of heat.

The larger the heating surfaces the greater will be the quantity of heat transferred to the water, which will be evaporated in larger quantity for the fuel burned, and the temperature of the terminal or waste gases will be correspondingly low. If a greater quantity of fuel be fired per square foot of heating surface, as would be the case if the boiler had a larger grate in proportion to the heating surfaces, then a greater volume of gas is formed relatively to the heating surfaces, so that each cubic foot or each pound of gas passes over a less amount, and so that while each square foot of heating surface absorbs a certain quantity of heat from the gases, the fact that there is more gas, volume or weight, relatively to heating surface, or, in other words, less heating surface in proportion to the gases available, means that the total heat is not reduced, as it would be had there been larger heat-absorbing surfaces, with the result that less water is evaporated for a given weight of fuel and the waste gas temperatures are higher.

The evaporation, however, per square foot of heating surface is considerable, owing to the larger amount of gas generated per square foot, and the consequently greater heat transfer per square foot of heating surface.

For any given rate of evaporation per square foot of heating surface, other things being the same, equal quantities of fuel will be required per square foot of heating surface; for instance, when boilers are compared on the basis of « equivalent evaporation », the coal required will be inversely proportional to the calorific value. If, therefore, one boiler has more grate surface than the other, then that having the larger grate will be required to burn a given quantity of fuel at a less rate, and on that account will be the more economical. For any boiler the rate of evaporation per pound of coal fired falls as the rate of combustion increases, not because the heating surfaces do not absorb a fixed certain amount of heat from the gases, but because, as the quantity of coal fired becomes larger, less of it is burned and a greater quantity is lost by sparks and imperfect combustion, owing chiefly to the difficulty of passing sufficient air through the fuel bed. In connection with the above figure 8 will be of in-

terest, as showing the relationship between the total coal fired per hour and the power developed. For the « Atlan-

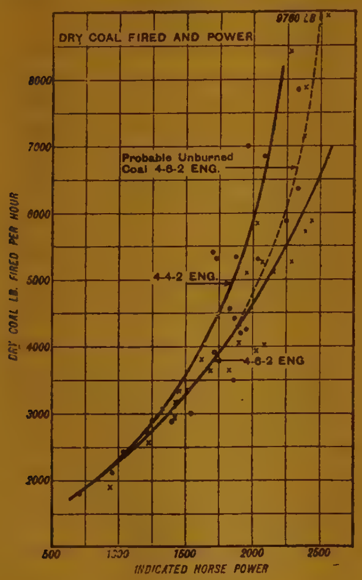


Fig. 8.

tic » engine a rapid increase in fuel consumption is indicated as the output of power increases. The « Pacific » type engine, on account of its more economical boiler performance, shows a more gradual increase in fuel consumption; but it is, however, evident that, at the higher rates of coal firing, evaporation and hence the power developed, did not increase; the fuel fired was not burned.

It would appear that when 8 000 lb. of coal were fired, or 150 lb. per square foot of grate per hour, there were very considerable losses through incomplete combustion. Figure 5 showed that evaporation begins to fail at that point, and figure 3 that the weight of sparks discharged reaches its maximum at about the same rate of firing. The temperature difference, however, between the fire-box and smoke-box is still about 1 700°, or what is was when the rate of combustion was only 75 lb. of dry coal per square foot or heating surface, which indicates, as has already been pointed out when discussing the fire-box and smoke-box temperatures, that each cubic foot or pound of gas liberated gives up a certain amount of heat to the heating surface. It therefore seems that failure to increase evaporation in proportion to a given increase in the rate of firing, is not due to the inability of the heating surfaces to absorb the additional heat available from the increased flow of gas, but rather to failure on the part of the fire-box to burn the coal fired effectually.

Long and short boiler tubes.

In connection with a study of the draught records shown by figures 1, 2 and 9 for any given rate of fuel consumption, the influence of the flue and tube length and also the gas area through the tubes and flues must be taken into consideration. From the particulars furnished of the two boilers under test it will be noticed that the outstanding difference is in the length of the tubes, those for the « Pacific » type locomotive being 21 feet long, as against 13 ft. 8 in. for the « Atlantic ». The ratio of proportion between the gas areas through the tubes and flues and the grate areas is approximately equal, from which it may be concluded that the greater draught necessary to produce a given rate of combustion in the case of the « Pacific » engine is due to the resistance to the passage of the products of combustion offered by the long tubes, which in this case have a ratio of length to inside diameter of 125, as against 94 for the « Atlantic ».

Figure 9 has been prepared to illustrate the comparative draught requirements at the smoke-box tube plate for the two boilers, one having long and the other short tubes. At low rates of combustion the difference is not very marked, but when an increase takes place considerable 1 700°, or what it was when the rate of

rate of 130 lb. the « Pacific » type boiler requires almost 2 inches more draught than the « Atlantic », whilst at 70 lb. per square foot of grate the difference is less than 1 inch. Further, for a given draught the « Atlantic » burns more fuel, the difference again increasing with the rate of combustion. Within certain limits, evaporation increases with the rate of combustion, so that for a given draught the « Atlantic » type boiler furnishes more steam per square foot of heating surface.

In figure 4 it was shown that at all rates of firing the temperature difference between the fire-box and smoke-box was greater for the « Pacific » type boiler having the longer tubes, and the evaporation performance has been seen to be better per pound of coal fired, and hence the boiler efficiency higher, more heat being absorbed. In deciding, then, how long tubes should be for a given locomotive having other related characteristics the same, the question to be answered appears to be how far or to what extent is economical steaming to be sacrificed in the interests of greater evaporation or free steaming. In this connection and in order that some idea may be formed as to the probable useful length for a tube of a given diameter figure 10 has been prepared to show the temperatures of the products of combustion as they flow through the tubes. In the tests the tube temperatures were obtained by a long thermo-couple inserted in a suitable tube 30 feet long, so that it would reach through the smoke-box and through the tubes to the fire-box end.

The couple was carried in the tube, the temperature of which was to be measured, by a star-shaped support, so that the couple was in the centre of the tube. The couple, after being inserted in the tube, was slowly withdrawn, the temperatures being taken at each foot of length of the tube. The flue tubes were treated in the same manner, but in this case the thermo-couple had to rest on the bottom of the flue on account of the elements. The temperatures, both in the tubes and flues, increased with the rate in firing, a condition which would be expected in view of the temperature records indicated by figure 4. The principal feature shown by figure 10 is the remarkable fall in the gas temperatures immediately on entering the tubes, and also the rapid fall in temperature during the first half of the length of each tube.

The curve for the first half falls quickly, and during the last half is very flat, indicating a much slower dissipation of the heat. The temperature in the smaller tube is 780° 12 ft. 11 in. from the fire-box. In the case shown for about equal quantities of coal fired the temperature difference is about equal, but from the temperature records shown by figure 4 it was seen that the smoke-box temperatures were lower by about 100° for the « Pacific » type boiler, the difference showing a tendency to increase at the higher rates of firing. It is evident from figure 10 that the temperature difference which takes place during the last 7 feet of the longer tube is very slight, apparently not more than about 50°, whilst from the run of the curves it would seem that the shorter tubes, if lengthened, might be more efficient in absorbing heat, for although the curve flattens considerably after the first half, there is a perceptible fall in temperature shown during the second half of their length.

Figure 10 shows the temperature condition for fuel consumptions of 5 888 lb. and 5 728 lb. per hour for the « Atlantic » and « Pacific » boilers, which correspond to firing rates of 104 lb. and 106 lb. per square foot of grate per hour, and from figure 9 it will be seen that the draught requirements are 7 inches and 7.9 inches respectively. The shortening of the tubes in the longer boiler would tend to reduce the resistance offered to the flow of the gases, and with the same draught lead to an increase in the rate of combustion, and because of that the some-

what reduced heating surface might give an equal total evaporation per hour.

There would, however, be some loss of heat due to increased terminal temperatures and a slightly lower boiler efficiency for a given rate of firing, because the longer tubes certainly effect a reduction in the smoke-box temperatures though their heat-absorbing value per unit of length is small and the value of the tube heating surface correspondingly reduced. On the other hand, shorter tubes allow of more fuel to be fired for a given draught, and the increased activity of combustion gives a greater evaporation per square foot of heating surface, a condition which is generally to be preferred in locomotive practice, where weight considerations per unit of power are usually of importance.

The temperature observations made showed that the particular position of the tube in the tube sheets made little or no difference to the temperatures; that is, tubes high up on the sheets showed much the same gas temperature as those set lower. The flue tube temperatures were for the « Atlantic » from 100° to 200° higher than those recorded in the tubes whilst those in the « Pacific » type boilers generally indicated lower gas temperatures by about 80°. The report expresses the view that these temperature readings may be in error, as great difficulty was experienced in obtaining definite readings, owing to « unstable conditions ». It seems reasonable to suppose that the gas temperatures might be higher in the larger tubes, owing in part to the larger volume of gas flowing through and passing over a hot tube surface, *viz.*, the superheater elements.

Superheater performance.

When the superheater forms an integral part of the boiler itself, as is the case with the Schmidt or flue-tube type as fitted to locomotive boilers, part of the heat units liberated by the fire-box are not available for evaporative purposes; they are absorbed by the heating surfaces formed by the elements comprising the superheater. The amount of heat taken up by the superheater depends upon the comparative volume of the gases which pass through the large tubes or flues, and the extent of the superheating surfaces and their position relative to the hottest part of the boiler which, in the case of the locomotive type of boiler, is the fire-box. Usually the elements are carried back to about 2 feet from the fire-box tube plate, and such was the case with the boilers now under consideration.

The evaporative performance of the boiler and superheater combined per square foot of heating surface per hour was shown for the two boilers by figure 5, and in figure 11 the equivalent evaporation of the boiler and superheater per square foot of heating surface is given separately plotted against the rate of firing dry coal per square foot per hour. The heat transfer across the water heating surface is proportional to the equivalent evaporation of the water, and that across the superheating surface is proportional to the weight of steam delivered to the engines and the heat units added during its passage through the elements, the equivalent evaporation of the superheater being proportional to the heat transfer thus obtained. The curves plotted, giving the relative evaporations, indicate the effectiveness of the evaporative and superheating surfaces in absorbing heat. The « Pacific » type boiler furnishes an equivalent evaporation per square foot of water heating surface of from 6.78 lb. to 17.62 lb. with a corresponding evaporation of from 1.82 lb. to 7.36 lb. of water for each square foot of superheating surface. The ratio of proportion of equivalent evaporation which takes place in the boiler and superheater varies from a minimum of 0.27 to a maximum of 0.42, showing that each square foot of heating surface in the superheater absorbs from 27 to 42 % of the amount

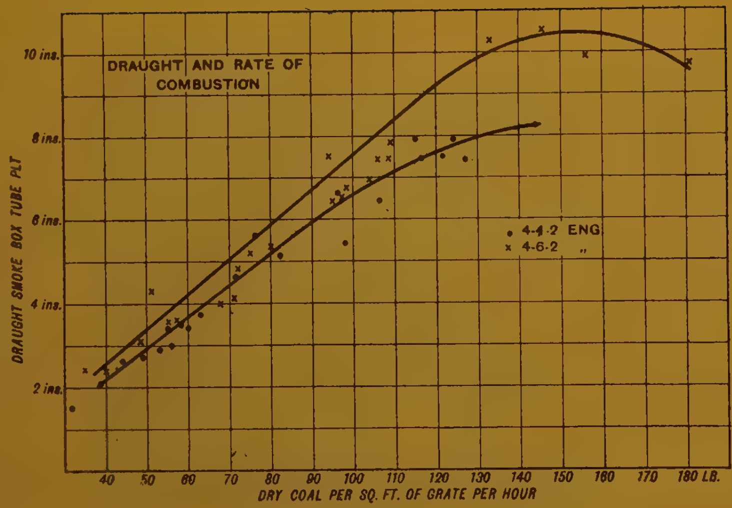


Fig. 9.

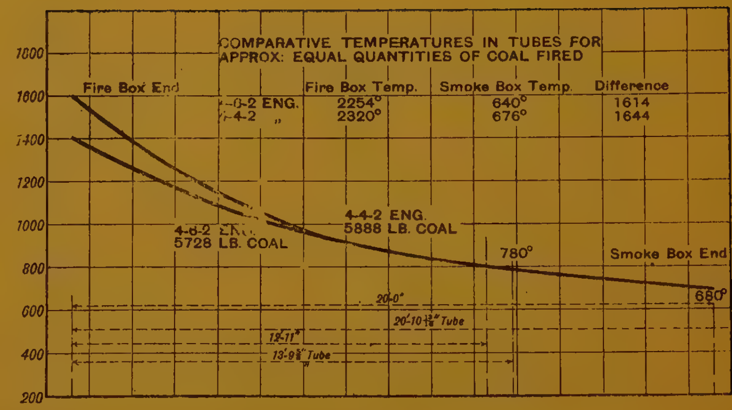


Fig. 10.

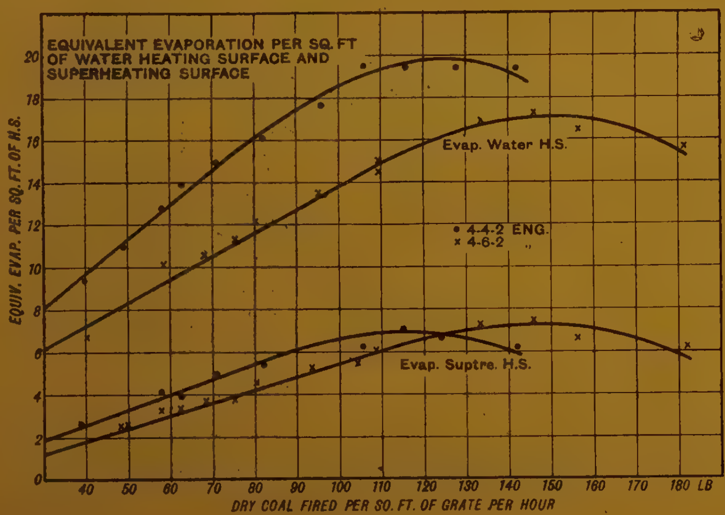


Fig. 11.

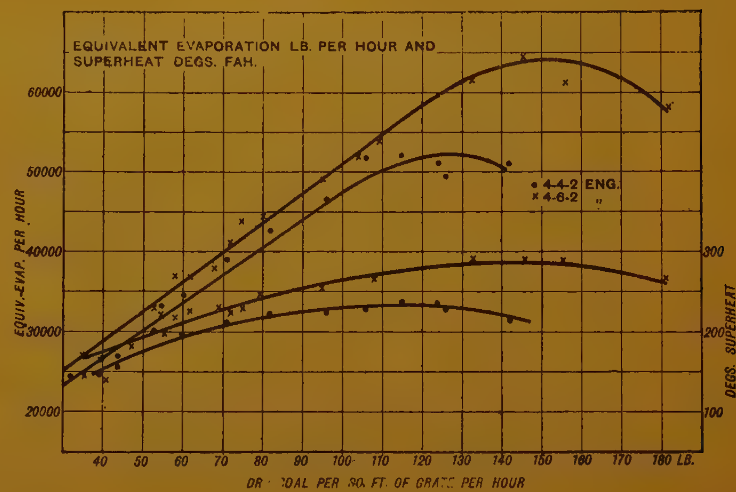


Fig. 12.

of heat taken up by a square foot of water heating surface, the average for these tests being about 36 %. In the « Atlantic » type boiler, the equivalent evaporation per square foot of water heating surface for which was from 8.42 lb. to 19.68 lb. per hour, the equivalent evaporation from the superheating surface was from 2.0 lb to 7.01 lb. In this case the ratio of equivalent evaporation in the boiler and superheater per square foot of heating surface varies from 0.237 to 0.356, or, in other words, the superheating surface absorbs from 23.7 to 35.6 % of the amount of heat absorbed by each square foot of water heating surface.

The absorption of heat by the steam passing through the superheater elements is not so rapid as by the water in the boiler. The heat transfer across the superheating surface per minute of the « Pacific » type locomotive varied from 28 854 to 133 740 British thermal units per minute and across the water heating surfaces from 364 243 to 920 583 British thermal units under maximum conditions, so the total heat transferred across the superheating surface was only about 14 % of the heat transferred across the water heating surface, although the superheater surface was nearly 30 % as large as the evaporative heating surface. The boiler on the « Atlantic » locomotive transferred from 21 259 to 79 513 British thermal units and from 328 917 to 764 269 British thermal units per minute for the superheating and evaporative heating surfaces respectively, so that for this boiler the rate of transfer across the superheating surface was about 10 % of the heat transfer across the water heating surface, the area of the superheating surface being about 25 % as large as the evaporative heating surface.

Figure 12 gives the equivalent hourly evaporation for each boiler at rates of firing from 40 lb. to, in the case of the « Atlantic » type boiler, 142 lb., and in the case of the « Pacific » 181 lb. of dry coal per hour, and also the degree of superheat obtained. The « Pacific » type boiler furnishes an equivalent evaporation of 64 700 lb. per hour, and at the same time the superheat attained is 280°, while the smaller boiler is seen to attain a maximum equivalent evaporation of 52 000 lb. per hour, the superheat at the same time being 230°, the curves showing the degrees of superheat, and, as mentioned when discussing figure 4 in connection with the fire-box and smoke-box temperatures, indicate that it remains practically constant at the higher rates of combustion and only falls off when the rate of evaporation begins to decline. The curves of superheat temperatures are, of course, exactly similar to those shown in the preceding figure, indicating the equivalent evaporation or the work done per square foot of superheating surface.

Figure 13 gives the total temperature of the superheated steam as delivered to the engines. They are similar to those in the preceding two figures, and are plotted against the rate of combustion. The temperature readings were taken from a point in the branch pipe between the superheater header and the steam chests. The degree of superheat follows the rate of evaporation and the large boiler evaporates the most water, while its superheater furnishes steam at a temperature which reaches a maximum of 670°. Between combustion rates of 90 lb. and 156 lb. of dry coal per square foot of grate per hour, the steam temperature is between 640° and 670°, during which the rate of evaporation is from 48 500 lb. to 64 000 lb. per hour.

Fuel and power developed.

Figure 14 shows the weight of steam delivered to the engines per hour and the power developed from it. The results for both locomotives are included in the plot and average lines drawn through the several points seem to show that the loco-

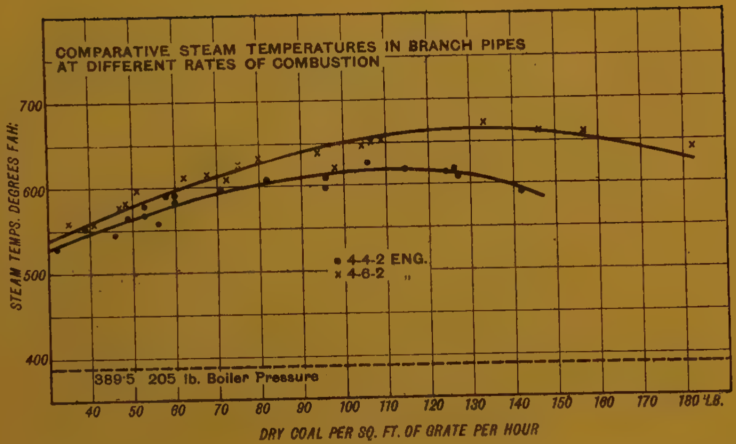


Fig. 13.

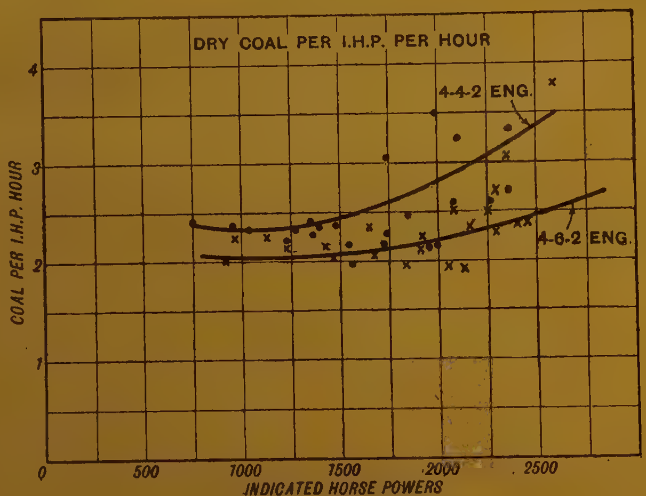


Fig. 15.

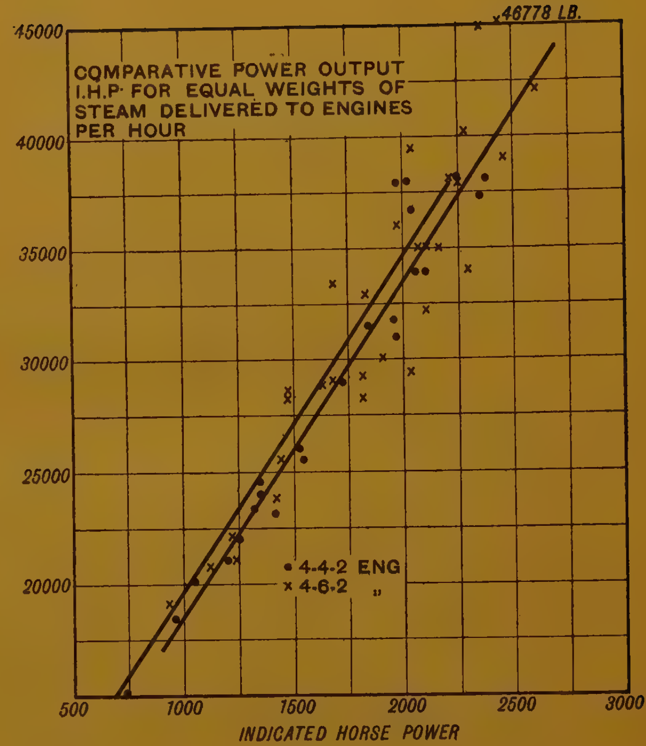


Fig. 14.

motives were, so far as the steam action in their cylinders is concerned, equally efficient. The fuel rate per indicated horse-power is given by figure 15, in which the dry coal per indicated horse-power is shown plotted against the indicated horse-power.

The results of the « Atlantic » type locomotive are somewhat widely separated, but the curve drawn appears to give a fair idea of the average results. In the case of the « Pacific » locomotive, the plotted points are more uniform in their position.

From what has been shown relative to the boiler performance of the two locomotives, and having in mind the results indicated by figure 14, the coal rate results exhibited by figure 15 are as would be expected. The « Pacific » type locomotive is the more economical of the two locomotives. Between 900 and 2 400 indicated horse-power, the coal consumption per horse-power hour is from 2.1 lb. to 2.4 lb. At 900 indicated horse-power the firing rate was 35 lb., and the water (equivalent evaporation) per pound of fuel was 12 lb., while at 2 400 indicated horse-power the rate of firing was 146.7 lb., and the equivalent evaporation per pound of coal 8.2 lb. The « Atlantic » locomotive operating between the same powers consumed from 2.4 lb. to 3.3 lb. of coal per horse-power hour, the firing rates being 39.2 lb. and 142 lb. of dry coal per hour, when the equivalent evaporations per pound of dry coal fired were 11.7 lb. to 6.6 lb.

It will be noticed that at the higher rates of power output the difference between the coal required per indicated horse-power increases. That for the « Atlantic » increases rapidly after about 1 700 indicated horse-power, and follows in this respect the relative positions taken by the lines giving the equivalent evaporation per pound of fuel fired, as shown in figure 6.

Conclusion.

Certain inferences may be drawn from a study of the more important actions which go to make up the functions of locomotive boilers, and may form a fitting conclusion to the present article. Those suggested are summarised as follows :

Blast and smoke-box conditions. — The importance of a correct form of blast orifice is established. The generally accepted circular form is not necessarily the best. Some type which will fill the chimney and at the same time entrain in its stream the exhaust gases is effective, and seems to indicate that an irregular form, as opposed to the circular type, might have considerable merit. The combustion possible depends on the draught or the smoke-box vacuum, which, in turn, depends upon the efficiency of the blast. There are considerable heat losses, through the throwing of sparks, which increases with the rate of steam production, owing to the greater draught action brought about by the increased quantity of steam discharged by the blast pipe. This, again, points to the importance of the form and size of the exhaust nozzle. While a study of the general performance characteristics shows beyond question how dependent are the various actions upon each other, it is possible that some form of blast nozzle which could be increased in size at high power output might offer advantage.

Temperatures and draught readings : Fire-box and smoke-box. — A boiler having a large grate area in proportion to the heating surface gives at all rates of combustion higher terminal or smoke-box temperatures than one having large heating surfaces and a comparatively small grate. For any boiler the temperature difference for all rates of firing is about the same. The gases liberated by the burning fuel, no matter how great their volume, are always reduced in tem-

peratures by a certain amount. The degree of temperature soon reaches a maximum value and remains almost constant throughout the higher rates of combustion. Fire-box draught readings increase with the rate of combustion, indicating greater resistance, caused, probably, by the thicker fire carried. The smoke-box draught increases with greater quantities of fuel fired, and shows continually increasing resistance which is due to the larger volume of gases to be moved through the tubes.

Tubes and flues. — For a given fire area through the tubes and flues and a given size of grate, long and short tubes each offer certain specific advantages. A long tube takes up more heat and is therefore more efficient so far as evaporation is concerned, but for a given rate of combustion requires more draught than a shorter tube. A short tube liberates more heat in the terminal gases, but for the same draught the rate of combustion is higher, and a greater evaporation is effected per square foot of heating surface. The fire burns better and the boiler steams more freely. The temperature drop in the tubes is remarkably rapid for approximately the first half of their length from the fire-box end. After that point, the gas temperatures fall more gradually until the curve of temperatures becomes almost flat, indicating a slow heat transfer. The best length of tube for any given diameter would seem to be that which is long enough under average draught conditions to permit of a reasonable absorption of heat. It should be of such a length that the curve of temperatures has just become flat before the gases are discharged into the smoke-box. Fuel economy is important, and so is free steaming, and the best length of tube is that which meets half way these conflicting conditions. The inside diameter multiplied by 100 seems to give a safe rule for the length in inches.

Evaporations and efficiencies. — For any given rate of combustion a boiler having a large grate and small heating surface evaporates more water per square foot of heating surface, and is smaller for a given power output than one having a smaller grate relative to the heating surface. For any given rate of evaporation, the boiler having the larger grate in relation to the heating surface is the more efficient, because for any evaporation the rate of firing is less per square foot of grate and the water evaporated per pound of fuel is greater. So far as the actual consumption of fuel and the process of evaporation are concerned, loss of efficiency is due to incomplete furnace conditions, and not to any failure on the part of the heating surfaces in absorbing heat from the gases available.

Superheaters and steam temperatures. — For any size of boiler the addition of a flue tube superheater reduces the area of the evaporative heating surfaces and also the amount of heat available for evaporation purposes. A boiler of the same size having equal amounts of heating surfaces supplying saturated steam, is more efficient than one of the same size and having the same amount of combined heating surface (water and superheating). It more effectually absorbs the heat available. This is due to the fact that the superheater-fitted boiler has less evaporative heating surface per square foot of grate, and also to the superheating surface being less efficient in the transfer of heat than is the water-heating surface. The degree of superheat imparted to the steam passing through the superheater increases with the rate of evaporation. As the evaporation increases with an increase in the volume of the gases forming the products of combustion, so does the superheat. This condition is further assisted by the increase in the relative effectiveness of the superheating surfaces as the rate of working increases.

Coal consumption and power. — The coal rate for any given power is dependent on the efficiency with which the boiler does its work, which consists of evaporating water. While in the present article no reference of any length has been purposely made to engine performance, it is known that the addition of the flue tube superheater giving high degrees of superheat to the steam has improved cylinder performance almost beyond recognition. So much so, in fact, that it seems that further fuel economies must be largely obtained by improved boiler operation, and the fire-box and grate seem to offer the most promising field for investigation.

[621 .335 (.494)]

The 1-C-1 (2-6-2) electric locomotives of the Swiss Federal Railways,

By E. SAVARY,

TRACTION ENGINEER OF THE SWISS FEDERAL RAILWAYS, LAUSANNE.

Figs. 1 to 4, p. 459.

(*Bulletin technique de la Suisse romande.*)

On the 13 December 1923, the first trials of electric single-phase traction were carried out with complete success on the first division of the Federal Railways. Single-phase electric traction is at present confined to the Sion-St. Maurice section (25.5 miles) and will reach Lausanne station this spring.

This section it will be remembered forms the continuation of the Iselle-Sion section (47 miles) which is electrified with three-phase current at 3 300 volts supplied by the hydro-electric power station at Massaboden, near Brigue.

The voltage of the single-phase current in the overhead contact wire is 15 000 volts at 16 2/3 periodicity. The current is supplied by the Barberine power station.

For working this service by single-phase electric current the first Division has at the present time a certain number of locomotives, built by the *Société anonyme des Ateliers de Sécheron* of Geneva, the mechanical details of which were supplied by the *Société suisse pour la construction de locomotives et de machines* of Winterthur. These locomotives are all Type A^e 3/5 (1-C-1) for express trains running on the flat country with six driving wheels and four carrying wheels. The following are their chief features (fig. 1):

Weight of locomotive in running order	61.1 t. (60 English tons.)
Adhesion weight	55.5 t. (54.5 English tons.)
Fixed wheel base	4.200 m. (13 ft. 9 5/8 in.)
Tractive effort for one hour at the wheel periphery	7.5 tons at 39 miles per hour.
Maximum tractive effort for starting	13.7 tons.
Horse power on one hour rating at the wheel periphery. . .	1 775 H. P. at 39 miles per hour.
Horse power on continuous rating	1 540 H. P. at 42 miles per hour.

Maximum speed	56 miles per hour.
Number of motors	3 × 2
Ratio of gear reduction	1 : 5
Electric brake	Not fitted.
Diameter of driving wheels	1 610 mm. (5 ft. 3 3/8 in.)
Diameter of carrying wheels	930 mm. (3 ft. 5/8 in.)

These locomotives, of which the mechanical arrangements are symmetrical, have an interesting peculiarity in the Westinghouse system of drive, applied to the three driving axles. Each of these axles is driven by a pair of traction motors coupled together with a common casing rigidly secured to the frame of the locomotive above the driving axle. The driving couple of each of the rotors is transmitted through a pinion to a toothed wheel carried on a hollow shaft concentric with the corresponding motor axle. The hollow shaft turns in bearings cast in one piece with the casing of the stators and arranged below the motors. Sufficient space is arranged between the hollow shaft and the driving axle that passes through it to allow for the vertical displacement of the axles. The driving couple from the hollow shaft is transmitted at each end of the hollow shaft to the corresponding wheel of the driving axle by an elastic arrangement consisting of six strong helical springs. This arrangement of springs forms on the one hand an elastic coupling for the transmission of the rotational movement of the hollow shaft to the driving axle, and on the other hand acts as a means of absorbing the shock caused by vertical forces between the axle and the motor arising from irregularities in the track. The elasticity of the coupling also acts to some extent in preventing slipping on starting (fig. 2).

The advantage of this system of independent drives for the driving axles is that it allows direct and continuous transmission of the driving couple without the reciprocating movement of the intermediate coupling rods. The arrangement and construction of the system also allows of increasing the breadth of the pole pieces of the motors (fig. 3).

Another advantage of this type of locomotive is found in the arrangement of the motors in three groups of coupled pairs. These groups are coupled in parallel and the two motors of each group are coupled in series. If one of the traction motors should become damaged, this arrangement enables the group, of which it forms part, to be put out of action by means of the corresponding main switch. The locomotive can then continue to run with a train load reduced by one-third. The load would be reduced by two-thirds if two groups of motors were damaged and cut out. We repeat that this is a valuable arrangement and one that is highly appreciated in service.

Each driver's cab, arranged at each end of the body of the locomotive, is fitted with the following mechanical brakes : a hand screw brake, the Westinghouse automatic brake, and the Westinghouse moderate application brake. Electric braking has not been fitted.

The current on the overhead contact line is collected by the locomotive by means of two pantograph collectors pneumatically operated from each of the driver's cabins. The high tension current at 15 000 volts passes from the pantographs in succession through section switches, a self-induction coil or other protection against excessive voltage, the maximum current relay, and the main switch to its connection to one end of the primary winding of a step transformer, the other end of which is con-

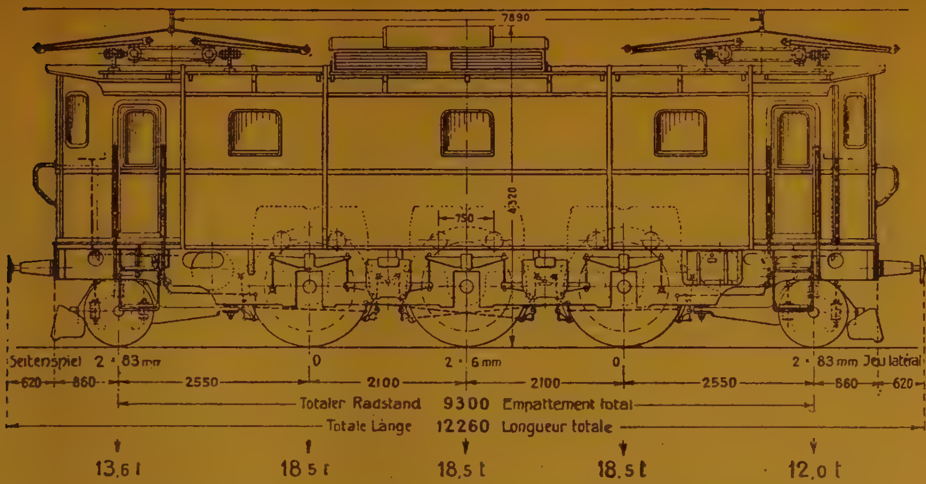


Fig. 1. — Diagrammatic elevation of the 1-C-1 electric locomotive.

Translation of French terms : Empattement total = Total wheelbase. — Longueur totale = Length overall. — Jeu lateral = Side play.

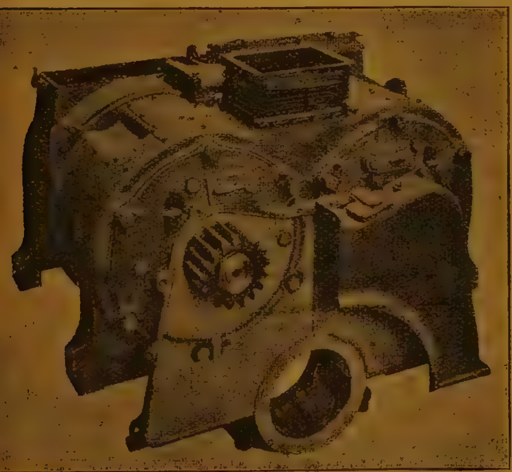


Fig. 2. — Twin traction motors.

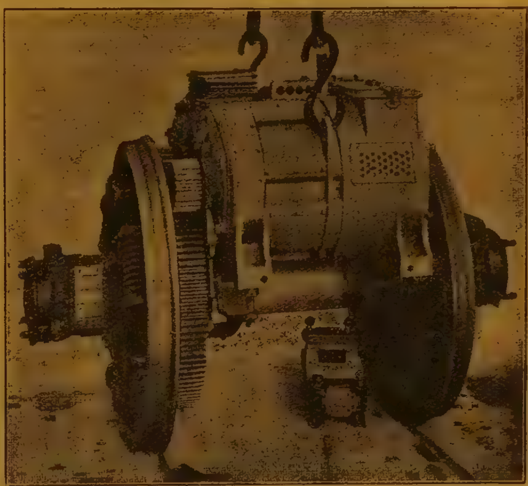


Fig. 3. — Traction motor as mounted in relation to the driving axle.

Fig. 4. — Battery of electro-pneumatic contactors.

nected to earth through the frame of the locomotive. The secondary winding of this transformer is arranged with nine contacts for current from 160 to 880 volts which supply the necessary current for the traction motors through batteries of contactors, reactance or regulating coils, maximum current relays and reversing switches.

The step transformer is of the oil-bath type cooled by two tubular radiators supplied with cold air by a fan.

For heating the train, the carriages of which are fitted for electric heating, three junction boxes are arranged on the secondary winding of the transformer at about 600, 800 and 1 000 volts. The current passes from one or the other of these junction boxes according to the amount of heating required, then through one of three regulating contactors, and a maximum current relay to the heating current couplings at one or other of the two ends of the locomotive, passing through the radiators arranged under the seats of the carriages.

The heating of the driver's cabs is fed from the 220-volt supply for the auxiliary machines.

The maximum current relays protect the circuit apparatus to which they belong against overloads and short-circuits by the automatic opening of the main oil switch by means of an energising current and a trip coil.

The no-voltage relay, arranged in one of the circuits of the cut-out coil of the main switch, throws open this switch when, for any reason the voltage falls or the pantographs leave the overhead line momentarily. The batteries of contactors are operated electro-pneumatically and also form one of the interesting features of the Sécheron locomotives. Each contactor consists of an air cylinder of which the distributing valve is controlled electro-magnetically from the driver's cab. These contactors, which are operated by means of air compressed to 5.5 and 7 atmospheres, regulate respectively the voltage of the traction and heating currents supplied to the train. For the traction current there are eighteen contactors giving eighteen steps of voltage for the supply to the motors under different conditions of load and train speed, and there are three reactance coils which limit the power interrupted by each contactor. The contactors are connected two by two to each of the nine contacts of the secondary winding of the transformer. The arrangement is such as to allow of progressive starting without shock (fig. 4).

The reversing switches for the traction motors are electro-pneumatically controlled, but they can also be operated directly by hand.

The driving controllers in the driver's cabs are each fitted with a handle for controlling the pantograph collectors, a handle for controlling the main oil switch, a handle controlling the reversing switches of the traction motors, and a handwheel for operating the contactors for regulating the supply to the motors according to the weight of the train and the speed required.

The control gear and contactors are interlocked so as to prevent any error of manipulation and to ensure their being worked in a pre-determined order.

Apart from the usual controller handle for locking and unlocking the main oil switch, the driver has, in each cab, an auxiliary cut-out arranged in the roof of the cab which he can use to ensure an emergency stop in case of danger.

Apart from the circuits for the traction motors and for heating the train which form the main circuits of the electrical equipment of the locomotive, there are secondary circuits for feeding the auxiliary machinery : the motor-fan, motor-compressor, motor-generator group and circuits for auxiliary purposes and lighting.

The motor-fan, the motor-compressor and the motor of the generator group are

supplied with current at 220 volts from the secondary-winding of the step transformer.

The two motor-fan groups, coupled in parallel, are controlled simultaneously from the driver's cabs. They ensure the cooling of the traction motors and of the step transformer.

The motor-compressor group can compress 70.63 cubic feet of air at atmospheric pressure per minute to a pressure of 7 atmospheres for feeding the two main reservoirs of the Westinghouse brakes, and of the pneumatic appliances. This group is also controlled from each driver's cab and the pressure in the reservoirs is regulated either by means of an automatic regulator or directly by hand.

The motor-generator group, also known as the convertor-group, produces continuous current at 45 volts for feeding the auxiliary circuits for working the auxiliary appliances and by means of an additional resistance, which reduces the voltage of the current from 45 to 36 volts, for the internal lighting of the locomotive and of the head lamps.

The circuit of the motor-generator group comprises also the groups of accumulator batteries coupled in parallel with the generator of the convertor group. These batteries, which are charged by this generator, supply current at 36 volts, and can, if required, supply it direct for a certain time to the circuits for auxiliary appliances and lighting.

Pneumatic and electric locking arrangements protect the staff against danger from high-tension current. Thus the door to the high-tension switchboard inside the locomotive can only be opened by means of a locking key which causes the pantographs to be lowered as soon as it is taken from its place. The opening of the high-tension door also ensures automatic earthing of the high-tension conductors on both sides of the main switch.

To obtain access to the roof the staff is supplied with a folding ladder attached to the outside of one of the sides of the end of the locomotive body. When this ladder is opened it causes an alarm whistle to blow and, at the same time, by reducing the air pressure in the pipe supplying the pantographs causes the latter to be lowered.

[625 .253]

A new Westinghouse air compressor for locomotives,

By J. NETTER.

Figs. 1 to 3, pp. 462 and 463.

(*Le Génie Civil.*)

The compressors at present in use on locomotives for providing air for braking trains work generally with steam at full pressure without cut off; they are therefore inefficient and are especially undesirable with the present cost of fuel.

This disadvantage is particularly serious in the case of suburban trains which make frequent stops, and with long freight trains, such as those which, under a recent order of the Allied Governments, were fitted with a continuous

brake on the lines of the Westinghouse brake [1]. For this reason the makers of this brake have themselves been considering a very powerful new compressor called the « bi-compound », which appears to us to be of such interest that we give a description of it, for apart from railway work, it appears to be adaptable to numerous uses. Fitted to a steam shovel, for example, it allows the economical compression of air for supplying various pneumatic tools, either for breaking up pieces which are too large, or those which are too heavy to be lifted in the shovel, or again, for drilling charging holes for blasting operations.

The new pump (figs. 1 to 3) is « compound » both for air and steam, that is to say, the compressing of the air is carried out in two stages, as in the « Fives-Lille » pump. With regard to the steam, after it has been used in a cylinder of small diameter, it is admitted to a large diameter cylinder, where it is again used before being exhausted to the atmosphere; its heat energy is therefore employed to the greatest extent, thus effecting great economy.

Comparative trials have shown that, for the same degree of compensation, the « bi-compound » pump shows a saving of 40 % of steam in comparison with the pumps at present in service, that is, taking coal at its present price, an annual saving of about 1 500 fr. per locomotive. It is generally admitted that a two stage pump consumes 25 kgr. of coal per hour, that is on an average 100 kgr. per day, whilst 100 kgr. of coal produces 700 kgr. of steam, this being the daily consumption for a two stage pump. With a « bi-compound » pump consuming only $700 \times \frac{60}{100} = 420$ kgr. of steam per day, a daily saving of 280 kgr. is made; this gives for 365 days a saving of 102 200 kgr. of steam, or $\frac{102\,200}{7} = 14.6$ t. of coal valued at 1 460 fr. taking the cost of coal at 100 fr. per tonne loaded on the tender.

(1) See, in this connection, the *Génie Civil* of the 27 January 1923 (t. LXXXII, No. 4, p. 86).

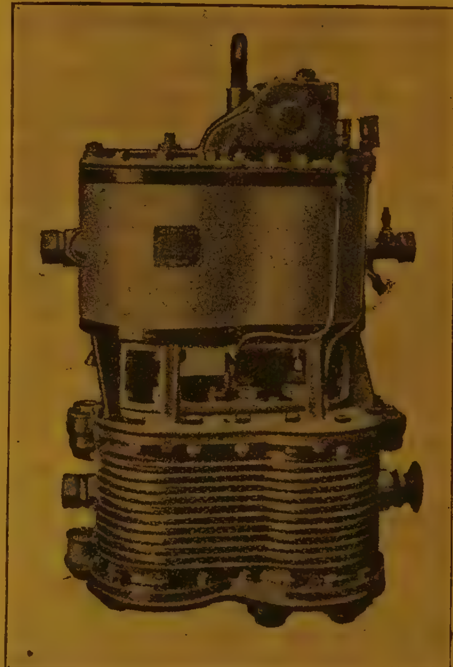
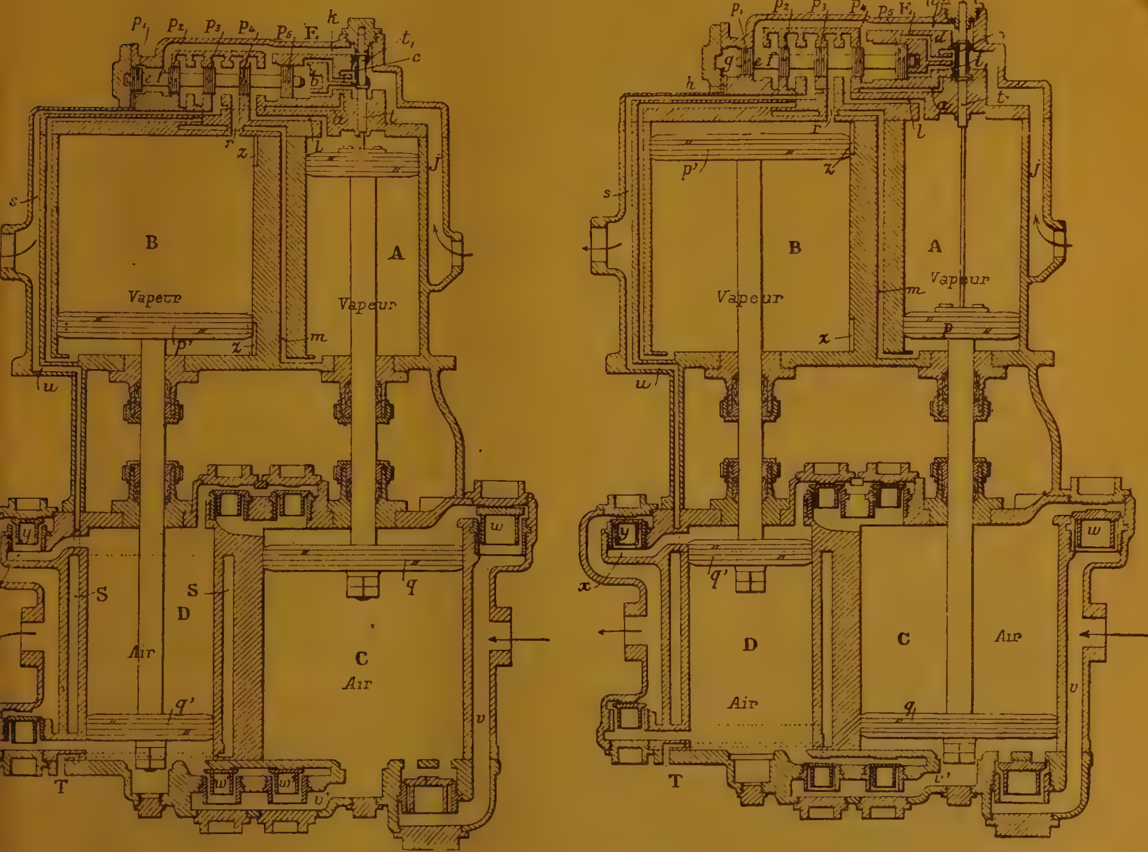


Fig. 1.
New Westinghouse air compressor.

Description of the mechanism. — The pump consists (figs. 2 and 3) of two steam cylinders A and B, two air cylinders C and D, and a steam distributor E. In order to make the method of working as clear as possible, the pipes and orifices have been shown diagrammatically without adhering to the actual method of construction, and the distributor E has been shown in a plane at right angles to its actual position.

The steam cylinders are placed above the air cylinders from which they are separated by an extension piece cast on to the steam cylinder, and which allows

of the adjustment of the glands. The high pressure steam cylinder A is mounted above the low pressure air cylinder C, so that the low pressure steam cylinder B is above the high pressure air cylinder D. The latter, in order to improve the cycle of compression, is cooled by the circulation of part of the exhaust steam through a jacket S cast interequally with the cylinder.



Figs. 2 and 3. — Sectional view of « bi-compound » Westinghouse air pump showing the pistons in their extreme positions.

The main valve. — The mechanism for distributing the steam consists essentially of a special differential piston valve p_1-p_5 forming a slide valve for distributing the steam, controlled by a reversing rod t and a reversing valve t_1. The control is operated by the piston p of the cylinder A each time it reaches the end of its stroke. It is thus given an alternating movement which it transfers to the differential piston valve because, dependent upon the position of the reversing valve, one of the faces of the differential piston valve is submitted

either to the pressure of the exhaust or the live steam on admission, the other faces of the differential piston valve being continually submitted to the same pressures.

For each extreme position of the differential piston, the intermediate distributing pistons establish communication between the various steam ports, so that the admission and exhaust of the steam for each cylinder is regulated correctly.

The operation of the distributor gear is as follows :

« The piston *p*, in completing its upward stroke, moves with it the reversing rod *t* and the reversing valve t_1. The steam then travels through the port in the chamber *b* and forces the differential piston p_1-p_5 to the left (fig. 2). In the same way, on reaching the bottom of its stroke, the piston *p* draws down the reversing rod and the reversing valve, thus connecting the chamber *b* with the steam exhaust through the port *c* of the slide valve and the pipe *d*. The differential piston valve is then brought over to the right, occupying the position shown in figure 3.

« The result is that the two internal faces *e* and *f* of the differential piston valve are continually under the pressure of the admission steam; the face *g* of the piston p_1 is always in contact with the exhaust steam through the port *h*, and when the face *b* of the piston p_5 is under the pressure of the exhaust, the differential piston valve is moved to the right because of the difference in the areas of pistons p_1 and p_5.

« The intermediate pistons p_2, p_3, p_4 are always balanced and do not exert any pressure in either direction. »

Operation. — This can easily be followed by referring to figures 2 and 3 respectively. The former shows admission of the steam to the top of the piston *p* in the cylinder A; the latter admission of the steam below the piston *p*.

The steam side. — Steam passes into the cylinder A through *j k l*, and acts on piston *p*. The steam which fills the cylinder under the piston *p* is admitted after expansion under the piston p' of the cylinder B by the passage *m*, and raises this piston. At the same time the steam above this piston p' exhausts into the atmosphere through the ports *r* and *s*; a portion of this exhaust steam passing through the port *u* into the jacket S and finally escaping into the atmosphere at T.

The air side. — The piston *q* of the air cylinder C is forced down by the piston *p*, and in so doing, air is drawn in on to the top of the piston *q* through the clack valve *w* and the port *v*. At the same time the air under this piston is compressed and driven through the port v' and the clack valves w' into the lower portion of the cylinder D.

The piston q' is operated by the steam piston p' and compresses the air in the cylinder D, and when the pressure reaches the predetermined point, the air is forced out through *x* and the clack valve *y*.

In the second phase of the operation (fig. 3), the action is analogous with that given above. When, however, the piston p' reaches the grooves z, the two sides of the piston are put into communication with one another and also connected with the exhaust through these grooves and the ports *r* and *s*. As the piston *p* does not meet with any resistance at the end of its stroke, it moves freely and so operates with certainty the reversal of the distributing valve.

[621 .138 (.73)]

Typical layouts for storage and distribution of fuel oil, including fuel oil stations between terminals.

Report of the Committee on Shops and Locomotive Terminals appointed by the American Railway Engineering Association.

(*Bulletin of the American Railway Engineering Association.*)

General.

The use of oil as fuel for locomotives is a development of the past twenty-five years, and has been widely extended since the war, as a result of the steadily rising cost of coal, coupled with largely increased production of residuum fuel oil as a by-product of the petroleum industry. Various elements in addition to the purchase cost per thermal unit enter into any consideration of the relative economy of oil and coal. Such are the saving in transportation cost due to the fact that the fuel value of oil per pound is about fifty per cent greater than that of bituminous coal; the reduction of waste and loss in handling and storing; the elimination of the soot and cinder nuisance and of damage due to spark-set fires; the possible increase in the length of engine runs and reduction in the number of stops required for receiving fuel and cleaning fires; and, on the other hand, increased cost of locomotive firebox and boiler maintenance due to the higher temperature of oil fires, and increased investment in facilities for handling and storing fuel.

A tabulation of replies from fourteen oil-using roads to a questionnaire covering general features of design of facilities for handling, storage and delivery of fuel oil is appended as information, as are a number of representative layout and detail plans of such facilities. Notwithstanding the comparatively recent development of the problem, and the varied conditions under which it has been worked out on the different roads, it is felt that these replies indicate sufficient agreement to permit formulating as recommended practice the general principles governing the design of such facilities.

There is a wide variation in the character of fuel oils, but the temperature at which the oil may be handled successfully and the extent of heating required to maintain that temperature are the main factors effecting variations in detail of design. The heavy Mexican crude or « topped » oil, with asphalt base and gravity as low as 10 to 12° on the Baume scale, is largely used by roads having access to gulf ports. This oil, when cold, is so thick and viscous as not to flow, and must be heated to temperature of 100 to 140° Fahr. to permit satisfactory handling. The lighter residuum fuel oil from the mid-continent fields, on the other hand, with paraffin base and gravity between 25 and 28° Baume, flows like water at ordinary temperature and requires heating only during extreme cold.

Oil is commonly delivered in tank cars for distribution to the various engine terminals. The Mexican oil is transported by tank steamers from Mexican ports to

the gulf ports and unloaded direct through pipe lines from the steamer to storage tanks. From the storage tanks it is pumped through loading racks to tank cars, as required for distribution. The domestic oil is delivered from the oil companies either direct to tank cars or companies either direct to tank cars or to conveniently located storage tanks

The facilities required by the railroads include provision for unloading the oil from the tank cars, for holding it in storage, and for delivering it to locomotive tenders, together with the necessary pumps and connecting pipe lines for transferring the oil between such facilities.

Unloading and pumping.

Oil is commonly unloaded from a tank car through the outlet, equipped with valves, in the bottom of the car. A pipe with swing joints or flexible hose may be connected to the outlet and oil delivered direct from the car to pump suction. This method is cumbersome and awkward to handle, and, in the frequent case where there is some defect in the valve, likely to result in loss of oil. A more satisfactory method, where considerable quantities of oil are to be handled, has been found to be to discharge freely into a box or trough placed between and below the level of the rails of the track on which the cars stand for unloading. A continuous steel trough, long enough to accommodate the number of cars to be unloaded at one time, and of varying dimensions to provide for flow to a central discharge pipe, gives satisfactory results. An alternative plan, reported in successful use by several roads, substitutes for the continuous through concrete or steel boxes about 8 feet long spaced at car length intervals and each provided with discharge pipe. Either trough or boxes may be equipped with metal covers, closed when not in use, and serving as windbreak to protect the discharging oil when opened for unloading. Typical plans for troughs and boxes are submitted as information.

From the unloading trough or boxes oil passes by gravity through discharge pipes to a depressed sump tank of steel or concrete. The questionnaire discloses wide variation in size and design of tanks in use for this purpose. They must be set low enough to provide for gravity discharge through the pipe line from trough or boxes as fast as oil is discharged from cars, and should be of sufficient capacity to receive in emergency a carload of oil when pumps are not in operation, or to allow time for closing of valves on cars in case of sudden stopping of pumps while unloading is in progress. Several roads report having experienced trouble with steel sump tanks being lifted out of place, when empty, through the action of ground or flood water. To avoid this, some method of anchoring such tanks with sufficient weight to overcome their buoyancy when empty is necessary.

For the lighter and more fluid oils, which require little or no heating, ordinary water pumps of various types are reported in successful use; with the heavier oils, the increased friction likely to be encountered may make desirable a pump designed for higher pressure and with special valves to provide free and unrestricted opening. When it becomes necessary to handle oil at high temperature, the use of metal valves and metal packing is common. No general conclusion can be drawn as to size and capacity of pump required, this being a matter to be determined for each individual case from the conditions to be met.

Storage.

Flat bottomed, cylindrical steel tanks, of sizes and designs standard with tank manufacturers for use by oil companies are in general use for storage. The size most commonly used for large storage

has a capacity of 55 000 barrels, and is about 115 feet in diameter by 30 feet in height. While some roads are using and others recommend all steel roofs for these storage tanks, the more general practice calls for wooden frame and sheathing, covered by light metal sheets, tar and gravel, or composition roofing.

As precaution against fire, storage tanks should be located as far from any other structures as can conveniently be arranged, consistent with economical handling and delivery of the oil. It is customary to surround each tank with an earthen dike or « fire wall » to retain the oil in case of fire or bursting of the tank. These fire walls enclose a volume from 20 to 50 % greater than the capacity of the tanks. Local regulations and fire underwriters' requirements should be considered in locating such facilities. Only two roads report the use of any protection against lightning.

The oil companies, which store large quantities of expensive oil, frequently install elaborate chemical fire fighting systems. None of the roads report the use of such installations, but most of them pipe steam to the top of the tank, with a view to smothering fires in the tank by discharging live steam over the surface of the oil.

To prevent dangerous accumulation of gases, it is customary to provide capped openings for ventilation in the roof, although some roads report that where wood roofs are used the unavoidable openings around the edges of such roofs provide ample ventilation. In general, it appears that the least ventilation and free movement of air over the surface of the oil consistent with safety is most desirable from the standpoint of losses due to evaporation. The following conclusions with reference to loss of oil by evaporation are quoted from a bulletin of the Bureau of Mines (Bulletin 200, Department of Interior, Bureau of Mines. Evaporation loss of petroleum in the mid-continent field, by J. H. Wiggins). While discussing primarily the losses from crude oil, the principles apply to evaporation loss for any fuel oil :

« 1° Evaporation during storage and handling causes one of the largest losses of crude petroleum between the well and the refinery;

« 2° From two-thirds to four-fifths of the evaporation loss may be eliminated by protecting oil from free contact with air. This protection will pay for itself in a short time;

« 3° The percentage of the original value lost by evaporation is two or three times the percentage of the original volume lost, because the fraction that escapes from the crude oil is the best gasoline and its value per unit of volume is two or three times that of the crude;

« 4° The maximum prevention of evaporation involves keeping the mixture of air and vapor above the oil at rest and as near a constant temperature, whether high or low, as possible. For example, if the mixture of air and vapor above the oil does not move, very little more evaporation will take place at 115° Fahr. than at 32° Fahr., because the mixture will soon become saturated and then evaporation will cease;

« 8° Overshot connections should never be used for filling a tank;

« 9° Even in winter, when atmospheric temperatures are low, oil in exposed lease tanks will lose more than one-half what it would lose during similar storage in summer;

« 10° Oil at the surface of 55 000-barrel tanks, where evaporation takes place, is subjected during a part of the day to temperatures much higher than the average temperature of all the oil in the tank. In summer these surface temperatures rise above 100° Fahr. Evidently in such storage the loss by evaporation must be large. »

Wide variations appear in the amount of storage considered necessary. Some

roads favorably situated as to source of supply require provision for less than a weeks consumption, while others recommend from three to six months' supply. A number provide storage for several months' requirements with a view to taking all possible advantage of varying market conditions. When central storage is provided from which oil is distributed in carloads to other terminals, it may be desirable to install loading racks, with movable pipe connections, spaced at car length intervals, for filling cars by pumping from storage or direct from elevated tanks.

Delivery.

Most of the roads prefer to deliver oil to the locomotives by gravity from an elevated tank. Several report using also direct pump or air pressure at small stations, or where local fire restrictions do not permit the use of elevated tanks. One only reports a preference for the use of pumps direct over the gravity system of delivery.

Various types and sizes of tank are in use for gravity delivery. Dismantled car tanks or similar horizontal steel cylinders mounted on steel or timber frame are in common use. In other cases, storage and delivery is combined in steel tanks of tower or standpipe type, with capacities of 100 000 or 150 000 gallons. The use of wooden tanks as containers for oil is generally considered not satisfactory.

The necessity for eliminating drip and wastage of oil makes the use of the ordinary water column for delivery to locomotives undesirable. The problem is to combine greatest possible flexibility of movement with oil-tight connections throughout and prevention of drip. Several types of columns are now manufactured for the purpose and are reported in satisfactory service by the different roads.

Two roads report the use of meters for measuring oil deliveries to locomotives. The majority, however, report fairly satisfactory results from the use of gauges to measure the oil in the tender after delivery.

Pipe Lines.

The economical size of pipe for handling oil depends upon the quantity which it is desired to handle in a given time, and this is affected somewhat by the character of the oil. In general, discharge lines from pumps are of six to eight-inch pipe, gravity delivery lines of eight to ten-inch. Pipes are commonly either buried in the ground or enclosed in a wooden or concrete box.

Heating.

Heating, where required, is by steam in all cases reported. Steam pipe coils are used in tanks. In the large storage tanks it is usual to enclose the coils in a wooden box located at the outlet, to get maximum benefit of the heat in the oil being pumped and avoid dissipation throughout the tank. When properly heated in tanks, several roads find it unnecessary to heat the oil pipe lines. The majority, however, provide for such heat, and practice reported and recommended is about equally divided between heating by a small steam line inside the large oil line and enclosing the steam line alongside the oil line in a box or conduit. The temperature to which oil must be heated to permit successful handling varies widely with the character of the oil and the climate, and the extent of provision required for heating varies accordingly. Two roads report successful use of thermostatic control for portions of their heating systems. The majority, however, have not used and do not recommend such installations, preferring to depend upon regulating valves operated by the man in charge of the station.

CONCLUSIONS.

General.

1. — Where oil is used as fuel for locomotives the facilities required include provision for unloading it from cars, for holding it in storage, and for delivering it to locomotive tenders.

2. — The details of design necessarily vary with the composition and gravity of the oil to be used and the climatic conditions to be encountered, as they affect the temperature which must be maintained in the oil for convenient handling.

Unloading Facilities.

3. — Oil should be unloaded from tank cars by discharging direct into a trough or boxes of steel or concrete between the rails of track on which cars stand for unloading. Where boxes are used, they should be spaced at car-length intervals for convenience in spotting cars for unloading.

4. — Unloading trough or boxes should deliver oil by gravity through pipe line to depressed sump from which it may be pumped to storage or delivery tank. Such pipe line should be of sufficient size and be laid with sufficient gradient so that oil will flow by gravity to the sump as fast as it will be discharged from the total number of cars which will be opened at any time. This should not be in excess of the capacity of the pumps.

5. — Sumps may be of steel or reinforced concrete and should be covered. They should have capacity of not less than one carload. If of steel, the pit should be drained or the sump should be anchored to prevent displacement by ground water when empty.

Storage.

6. — The storage capacity which should be provided depends largely upon reliability and source of supply and probable variations in market price of oil. In general, there should be at each station sufficient storage to protect against any interruption which may occur in the delivery from the regular source of supply. Additional storage for the purpose of taking advantage of variations in market conditions may either be located at various terminals where oil is used, or concentrated at one conveniently located point.

7. — Cylindrical steel tanks of 55 000 and 80 000 barrels capacity, erected on leveled earth foundations, provide convenient and economical storage, and can commonly be secured promptly and at less cost on account of being standard construction with tank manufacturers. Each tank should be surrounded by an earth dike, enclosing below the elevation of top of dike a volume equal to one and one-half times the capacity of the tank. Roofs should be provided of steel or of wooden frame and sheathing, covered with asbestos, composition, tar and gravel, or sheet metal roofing.

8. — Adequate means should be provided for the escape of gases thrown off from the surface of the oil. The character and extent of such provision required will depend on the tightness of the roof and the character of the oil. It should be designed to reduce circulation of air over the surface of the oil to a minimum consistent with prevention of building up of pressure due to the accumulation of gases.

9. — Provision should be made for draining off water and refuse which may settle in the bottom of tanks.

Delivery

10. — Oil may be delivered to locomotive tenders by gravity from elevated steel tanks or under direct pump pressure. In general, the former method is more convenient and economical.

11. — The size of delivery tank required varies with local conditions as to receipt and handling of oil, but the capacity should, in general, be not less than

the average amount of oil to be delivered in twenty-four hours.

12. — Valves should be provided for draining off water and refuse which may accumulate in the bottom of tanks.

13. — Delivery columns should be so constructed that spout can be swung to position and valve opened from the locomotive tender to be served. Spouts should have maximum freedom of movement in both horizontal and vertical directions, consistent with prevention of leakage. They should be provided with drip bucket, reversible end elbow, or other means to prevent drip.

14. — Means should be provided for measuring accurately deliveries of oil. Meters in delivery pipe lines or gauges on engine tenders serve satisfactorily to that end.

15. — Some wastage of oil around an engine terminal is inevitable and provision which will reduce such wastage to a minimum is an important item in design of facilities for handling oil. If all unnecessary waste and leakage is eliminated the cost of recovery of waste oil is generally in excess of the value of the oil. In cases where such waste is excessive or becomes a nuisance, however, and causes damage to neighboring property, it becomes necessary to provide traps in drainage channels or sewers, equipped with baffles, to cach the waste oil, separate it from water, and permit its recovery by dipping or pumping back to the sump. Such appliances are being used successfully.

Heating.

16. — Where heavy oil is used or where cold temperatures are experienced, it is necessary to provide means for heating oil in cars, tanks and pipe lines, in order that it may flow freely. Such heat is best provided by steam pipes.

17. — Pipe coils in tank cars, which can be readily connected by flexible hose or pipe to steam pipe lines from the pump house, provide satisfactory means for heating before unloading. The discharge of live steam directly into the oil in the car may be resorted to in case heating coils are out of order or car is not equipped.

18. — Similar steam pipe coils provide satisfactory heat for storage and delivery tanks. In larger tanks they are more effective if enclosed with the end of the discharge line leading from the tank in a wood box so that the heat will be applied directly to the oil as it leaves the tank, and not disseminated through the whole tank full of oil. The heating of oil in pipe lines will often prove advantageous and may be accomplished by introduction of small steam pipe lines inside the oil lines, or by enclosing steam line inside an insulating box alongside the oil line. The latter method simplifies construction and maintenance, but requires more expensive first installation and greater consumption of steam in proportion to the results obtained.

19. — Where steam lines are installed in oil lines, it is necessary to take precaution against excessive heating. On this account, it is not recommended that steam lines be so installed larger than necessary for heating the pipe line. Steam for tank coils and other purposes may better be carried outside the oil lines.

Small stations.

20. — While the foregoing recommendations apply primarily to the larger stations, yet the general principles apply to the small stations except that their application requires special adaptation to the problem. In some cases, the oil is used direct from the cars, in other cases, storage from one or more cars is combined with delivery tanks, delivery being made either by gravity, pumps or air pressure.

[656. 254 (.73)]

Automatic signals for a railroad grade crossing.

Figs. 1 and 2, pp. 471 and 473.

(From the *Railway Age*.)

The New York Central, with the approval of the Canadian National Railways and with the approval of the New York State Public Service Commission, has arranged for the installation of automatic railroad grade crossing signals at Helena, St. Lawrence County, N. Y., where the New York Central's line to Ottawa, Ont., crosses the Canadian National (the Massena branch of the Grand Trunk). This plant is here described.

The Pennsylvania has agreed with the New York Central for the substitution, as soon as plans have been prepared, of similar crossing signals for the mechanical interlocking now in use at the crossing of the two roads at Madera, Pa. Madera is in the wilds of the coal regions, 20 miles north of Altoona.

In view of the greater reliability of colored light signals as compared with motor-operated semaphores, and the elimination of moving parts which may cause false clear indications, the New York Central has decided that for future installations of automatic railroad grade crossing signals colored light signals shall be used. This type of signal is, therefore, to be put in at Helena. It is the Hall single-light signal, with movable roundels, described in the *Railway Age*, 14 January 1922, page 186. The appearance of a single signal is illustrated herewith, figure 1. (At Helena, double signals are used, as indicated in the drawing).

As light signals lack the convenient circuit controllers of the semaphore apparatus, the circuits for Helena are materially different from those, where semaphores are used. The control is as flexible and as reliable as that obtained

Fig. 1. — Hall « Searchlight » signal.

with the semaphore and there will be more efficient working. With light signals, the only moving parts will be those of the relays and the electro magnets of the signal mechanisms; and with current supplied from 1 000-am-

pere-hour primary batteries, the arrangement should be safe, reliable, efficient and economical; and easy to maintain.

A typical plan of the arrangement of signals to be installed at Helena is shown in figure 2, for which we are indebted to W. H. Elliott, signal engineer of the New York Central. As a complete circuit plan is somewhat difficult to follow, this is a partial plan, designated to show the typical and principal controls to be provided. It is to be noted that signals of two lights (home signals) are provided to govern movements over the crossing (signals 1, 2, 3, 4) and each home signal has an approach signal. The approach signals (651, 636, 5, 6) are threeindirection. The home signal has but two indications, displaying green or red above and yellow or red below. Marker lights (red) for the approach signals, although shown on the plan, will not be used on the Canadian National signals.

As noted, this is not a complete plan; and the tracks are not drawn to scale.

The circuits are arranged to permit the first train coming on to an approach lighting or preliminary clearing circuit, to proceed over the crossing under clear signals; and by its presence to arrange the circuits for the other road so that the signals on that road will indicate (provided a train approaching on that road has entered the approach lighting circuit, making it possible to energize the signal lamps) : at the distant signal, *caution* — approach next signal prepared to stop — and at the home signal, *stop*.

When no train is *approaching* the crossing, all lights are de-energized — dead. Even with a locomotive or a car standing on the crossing, and by its presence there making the electrical contacts at the signals of the other road necessary to protect itself from trains on that road, the signals of the other road do not light up until an approaching train needs them.

Should two trains on the same road approach the crossing running from opposite directions, the first train to run on an approach lighting circuit will cause clear signals to be displayed for itself — green by the upper light on the approach signal and green by the upper light on the home signal. Then, when the train moving in the oposite direction runs on the approach lighting circuit on its side of the crossing, it will cause signals to be displayed for itself as follows : in the approach signal a yellow over a red light, and in the home signal a red over a yellow light; and the signals for the other train — the one which approached first — will also be made to display these same indications. Thus each train, as soon as two are approaching together, from opposite directions, sets a caution (slowspeed) signal against the other one. These indications having been displayed, switching movements may be made in either direction, on that one line, over the crossing, without flagging.

If any of the track circuits of one road are occupied by a train, the signals of the transverse road will be made to display proper indications to cause a train on that road to stop at the crossing; and the train cannot get a signal to proceed until the train on the road first occupied has gone over the crossing and its rear end has cleared the crossing track circuit. On the rear end so clearing, the home signal of the other road will at once indicate « Proceed, » to permit the other train to proceed over the crossing. As long as the crossing track circuit (which is only about 150 feet long), or the track circuit for an approach to the crossing, is occupied by a train, the route is held for that train. The train gives up the route only when its rear end has passed over the crossing and it is clear of the crossing track circuit.

The circuit have been arranged to provide direct control by relay for each signal indication, except where it is necessary that control be had through the contacts of the signal relay mechanism. The mechanism contacts are operated by a cam on the armature shaft, according

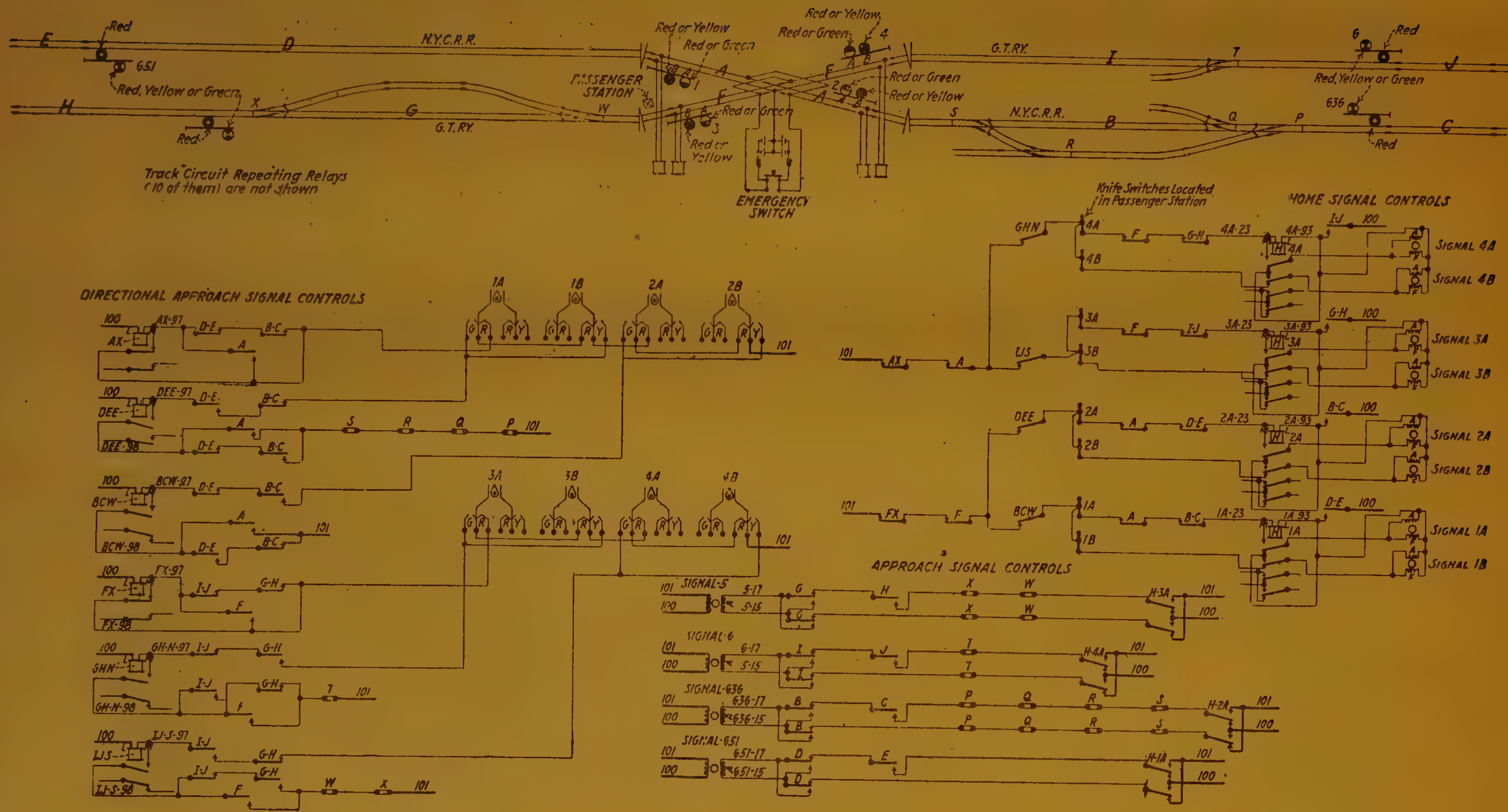


Fig. 2. — Automatic signals for a grade crossing, Helena, N. Y. Typical circuits.

to the direction in which the armature is turned, to cause a yellow or a green light to be displayed. These contacts on the signal mechanism are the means by which signals of the transverse road are held at stop, when either of the home signals of the road first occupied indicates proceed.

Track circuits are of the end fed type, except « A » and « F » which are center fed having a relay at each end. The track battery is controlled through a knife switch located conveniently at the crossing. In accordance with a card of instructions this knife switch can be used, in case of signal failure, by the conductor of any train which cannot get a proceed signal, to set signals on the other road at stop, so as to permit his own train to proceed over the crossing.

Relay AX is of the stick type, the pick-up of which is controlled through the front contact of track relays « D », « E », « B » and « C » and through contacts on signals 1A, 1B, 2A and 2B, made only when the signal displays a red indication. The control through track circuits « D », « E », « B » and « C » is shunted out through a back contact on track circuit « A ». The purpose of this relay is to hold the crossing signals on the Canadian National in the stop position, when a train is approaching the crossing in either direction on the New York Central, and the N. Y. C. signal displays an indication to proceed over crossing.

DEE relay is of the stick type, normally de-energized. The pick-up is controlled through the back contact of the D-E relay, through front contact of B-C relay, through circuit controller on signals 1A and 1B in multiple, made only when signals display green or yellow. The circuit holding the relay energized is controlled through front contact of D-E relay, back contact of B-C relay and through switch circuit controllers on switches « S », « R », « Q » and « P ». The control through relays D-E and B-C is shunted out through a back contact on the track relay for Section A. The control provided by these relays is designed to release the route for a train to pass on the transverse road when the rear of a train on the road first occupied passes over and off of the short crossing track circuit, it being necessary, in order to accomplish this, to keep the signal on the first road from clearing for a movement in the opposite direction.

Signals No. 5 and No. 6, and signals 636 and 651 are the approach signals for trains coming from, respectively, Massena, Montreal, Utica and Ottawa. Energy is applied to the field side of the mechanism at all times. For the control on the armature side, energy is taken through pole-changing contacts on the H relays for the home signals of the tracks governed. The control is then through all switch circuit controllers and track relays between the approach and the home signals and through the approach lighting relays. When a home relay is de-energized, current flows in the direction causing the approach signal to display yellow. When a home relay is energized, current flows in the opposite direction causing the approach signal to display green.

Signals 1A, 2A, 3A and 4A are home signals. The operating coils of these signals are controlled through H relay, and through all track circuits and switches in advance of the signal, including the approach clearing circuits in the opposite direction; through track circuits between the home signals on the transverse road, through a knife switch in the passenger station at the crossing and through back contacts of stick relays, « BCW », « DEE », « IJS » or « GHN » respectively. Signals 1B, 2B, 3B and 4B are controlled through back contact of H relays, through knife switches at passenger station, through back contact of « BCW », « DEE », « IJS » and « GHN » respectively; through the track circuit of the transverse road between home signals, and through AX or FX relays. The armature and field coils for each

home signal are in multiple. When H relay is de-energized, a shunt is provided on « A » signal through back contact. A shunt is provided for « B » signal through front contact of H relay. The H relays are normally de-energized, being energized through back contact of track-circuit repeating relays, when a train enters all approach clearing section.

The home signals in addition to the automatic controls, are controlled by knife switches, operated by the station agent for manual block operation. A knife switch is provided for each home signal, so that when the block extending to the next station is unoccupied a train may be allowed to pass into the block on a green indication of the top signal. When the block is occupied a train may be permitted to enter the block on a yellow indication of the lower signal. Thus by the use of the knife switches the station agent has a complete block signaling outfit. A train for Ottawa, for example, having departed from the station, signal A2, by means of the knife switch, is held at stop (red) until the train reaches the next station.

Track relays are to be of the Hall gauze contact type with batteries of 1 000 ampere-hour caustic soda type. Signals are to be of the three-position Hall color-light type, having armature resistance of 250 ohms and 500-ohm field coils.

If a signal governing a movement over the crossing does not clear when it should, it is necessary for a trainman to go to the crossing and, after ascertaining that no trains are approaching, unlock a box and open the two-way single-throw knife switch which cuts the current off from the two track circuits that cover the crossing thus setting at stop the controlling signals on both roads; after which the train may be moved over the crossing under protection of a flag. The bulletin prescribing the practice to be followed in such a case, is worded as follows :

« A signal indicating stop may be passed only on hand signal from trainman standing on crossing. Trainman must, before giving hand signal, observe that no train is approaching on any track, and then only after he has unlocked box at crossing and opened crossing switch. After train passes, crossing switch must be closed and box locked. »

The same process may be followed in case a train on the other road, having the right to the crossing, is standing at the home signal, and is willing to relinquish its right.

As it is desirable that the track circuits of the crossing be as reliable in their working as it is practicable to make them, the home signals are placed as close to the fouling point at the crossing as the local conditions as to rail joints, switches, platforms, etc., will permit, 50 feet being prescribed as the minimum distance. With the signals so located and derails not being used, the crossing track circuits are of a minimum length; and with relays at each end of each circuit there is small chance that the relays will fail to operate to cause the home signals to indicate stop whenever the crossing is occupied.

As the approach signals are located braking distance from the home signals for the maximum speeds authorized on these divisions, trains receiving proper signal indications will not be required to reduce speed below that prescribed for a crossing protected by interlocking signals. From an operating standpoint the arrangement is as efficient for through movements as any other system of signals; and where switching movements are to be made, within the track circuit limits, it provides block signal protection with a promptness in the display of signal indications that may be had only from apparatus that is automatic in operation.

For the benefit of the reader who is not familiar with automatic signals it may be explained that when a train or any part of it is moving or standing on

a section of track, in the rails of which the electric current of the signal system is flowing, the signals which notify all other trains to keep off that section of track are automatically set at « stop ». The presence of the wheels of the cars de-energizes one or more electro-magnets; and the electro-magnets act to move all signals — one or many — on all conflicting tracks or routes to indicate « stop » to approaching trains.

The light signals are electric lights, properly hooded, so that they are seen by the engineman in daylight as well as at night; and in this case the circuits controlling them are extended to a point about a mile back of the approach signal and there controlled by a track relay so that the lamps are *never lighted* except when a train comes on to the rails of the lighting section. Thus each approaching train lights its own signals a proper distance ahead though, of course, these signals are potentially energized by the absence of any train of the transverse railroad on or approaching the crossing.

In the drawing the track circuit limits are indicated by thick spots in the lines which show the rails.

The color-light indications are : red for stop, yellow for caution and green for proceed.

The Hall light signals differ from other light signals in that there is only one lamp for all three colors, an armature of very light weight serving to change the colors by moving small, thin roundels to position in front of the lamp. The roundels are 1 inch in diameter and 1/16 inch thick.

[656 .25 (04 (.42)]

The profession of the railway signal and telegraph engineer, (1)

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SIGNAL AND TELEGRAPH SUPERINTENDENT, SOUTHERN RAILWAY.

(*The Railway Gazette.*)

The railway signal and telegraph profession has a very good record behind it, and, although it has recently been stated that there is little opening in Great Britain for mechanical signalling engineering, I venture to predict an increased and useful future, not only for mechanical signalling, but for signalling and telegraph engineering in this country, and it will be the pleasurable duty of this Institution to use its efforts to that end.

The profession is by no means as young as is believed by those outside of it. Its importance and advantages have not, in recent years, been in my opinion recognised by railway authorities generally in this country, not only its past and present services, but its usefulness and capability of rendering more important service in the future, in development of the railways of this country and increasing their efficiency. Signal engineers will, I am sure, have wide scope to apply the principles of signalling to the problems of railway transportation, and to the increased capacity of the railway lines with advantage. There are several reasons for this want of appreciation of

(1) Abstract of the Presidential Address delivered at the Institution of Railway Signal Engineers, on 13 February 1924.

the good work of English railway signalling.

I will take the telegraph part of our work first. It is a fact that the first telegraph engineers (I include in this, contractors as well as railway telegraph engineers) were railway telegraph engineers. In 1840 to 1845, many wires were being run along railways. This was long before there was any idea of the Post Office telegraph engineer, who came along with the Telegraph Act of 1868. Railway telegraph engineers were pioneers in this branch. When the Post Office took over the monopoly of the telegraphs, the profession split into railway and Post Office engineers. The railway engineers progressed along the line of block, lock and block, tablet, electric staff, railway telegraph and telephones. The Post Office engineers kept more nearly to telegraph and telephone work, and mighty strides they have made, but it must not be forgotten that their base was the railway telegraph engineer, and the old electric telegraph companies' engineers before 1868. None the less progress has been made by the railway telegraph engineers in the branch of the work they took up. The installation of block signalling generally on sound principles on the 25 000 miles of railway in Great Britain and Ireland is a creditable achievement. Then the development of lock and block, or the interlocking of the mechanical with the electric block signalling of various types, track circuits, and various other systems of safety appliances electric fouling bars, train describers, and later, power signalling. The telegraph system most suitable for railway working and the various telephones that have been introduced on railways show that they have not been pioneers only in early days, but they have been pioneers all along. They have been the first to try out and introduce different devices to see if they are suitable for successful and safe railway working.

It is of interest to remember that Mr. W. Langdon, a railway telegraph engineer, tried an automatic warning to the drivers passing a signal at danger at Gunnersbury on the London & South Western Railway about 1868, and that it was he who first propounded a practical scheme for electric traction on railways to the Institute of Electrical Engineers, the principles of which are in use on a large proportion of the electric railways in this country to-day. He proposed 500 volts on the trains, but high voltage transmission.

The railway telegraph engineers were amongst the first members of the Institute of Electrical Engineers which catered for them very well in the early days, but railway telegraphs have become so specialised, and they (Institute of Electrical Engineers) are so taken up with heavy electrical engineering that there has been of late very little attention given to railway electric telegraph subjects — hence one of the needs for this Institution.

Signalling is nearly as old as railways in one way or another. The first signal I can find tidings of was on the Liverpool and Manchester Railway in 1834. Sir John Wolfe Barry said that the types of fixed signals originally used on the various railways in England differed according to each engineer. The introduction of the semaphore arm by Mr. C. H. Gregory altered this for daylight signals, and it was very soon universally in use. At night time red for danger and green for clear have been in use from early times.

There is a very large field and scope for our Institution, and its influence will grow as time goes on, if it applies itself to the problems of the day, but this means a real effort on the part of every member, for although we wish to benefit the whole of the profession our real duty is to look after the interests of the railway companies' work. We may not be able to enforce our recommendations, but the many questions that arise, and the problems

that come up, can be discussed in open session, and such will influence public opinion of railways very much.

This has been the case with the Railway Signal Association of America, now the Signal Section of the American Railway Association. They have produced standard specifications and drawings for a very large number of articles, apparatus, etc., in use in signalling practice. It will be seen, from a national or imperial point of view, that this is of great importance to the manufacturers of signalling materials in this country, and secondary to the railway companies also, as the price of apparatus depends largely upon the quantity of materials made in a given time. There should be no reason why the Institution should not get out specifications, etc., or combine with the British Engineering Standardisation Association to bring about standardisation in regard to materials in which they are specially interested.

It is difficult to show in actual money the value of the signal and telegraph engineer's work, because the accidents prevented by the signalling appliances cannot be calculated with any degree of accuracy. The value of keeping trains in motion, or the value of a telephone circuit in the additional facilities, both for commercial and operating purposes, the value of the time saved through the signalmen's use of the telephone for controlling traffic, is very high, but it cannot be calculated. The signalmen could not deal with the traffic of to-day without the telephone. It would be difficult to say in exact figures the value of the signals in the safe movements of traffic.

The yearly financial statements of the various companies show what amounts have been expended on the maintenance of signals and telegraphs. I have prepared the accompanying table for the year 1922 — the last of the old companies.

The modern work of the signal engineer is to keep the traffic moving at the highest speed with safety and to prevent stops, if possible, and to increase the capacity of the railway lines to a maximum, where wanted, at the lowest cost, efficiently and economically, and to give the necessary freedom of movements at stations with the standard degree of safety.

The advantages that modern signalling can give to railway working are very much more appreciated in America and many of the Colonial Dominions than, generally, in Great Britain to-day. Greater use is being made of the modern practice of signalling abroad than here, and they reap a correspondingly greater efficiency from it.

Generally speaking, the importance of seconds or even minutes on the working of the traffic is not realised by the ordinary railwayman. How, at times, the quicker movement of a piece of apparatus by 3 or 4 seconds would prevent a train stopping is not considered — the 3 or 4 seconds is thought to be too small to count.

Now as to the future. Gratitude is said to be the expectation of favours to come. What advantages can the signalling profession offer to railways in Great Britain to assist in the economical working of traffic?

1° The railway signal and telegraph engineer has an excellent opportunity to so manipulate the telegraph wires on any system by well-known means of superimposing and other means of a like character, so as to give improved telephone communication between stations and, if necessary, telegraph at the same time as well. Automatic telephone system can be arranged to give an automatic calling system to cover a number of offices and stations in a local, but very wide area, which dispenses to a large extent with operators, but to make it pay it must be properly arranged and for comprehensive operation;

2° Working several existing block sections from one signal-box;

Comparative statement of the cost of maintaining the signals and telegraphs on 20 typical railways in Great Britain for the year ended 31 December 1922.

COMPANY.	Signal maintenance.	Telegraph maintenance.	Total maintenance signal and telegraph.	No. of route miles of railway.	No. of single-line track running lines.	Maintenance cost of signal and telegraph per route mile.	Maintenance cost of signal and telegraph per single mile of track running.	Total expenditure on railway.	Per cent of total expenditure spent on signal and telegraph.	Total expenditure on signalmen and gatemen.	Cost per route mile of railway of signalmen and gatemen.
	£	£	£			£	£	£		£	£
London & South Western Railway.	113 186	54 623	167 809	1 019.18	1 895.0	164.6	88.56	9 183 932	1.82	231 132	226.82
London Brighton & South Coast Railway.	65 116	40 885	106 001	457.22	910.77	231.94	116.48	5 349 242	1.963	165 640	362.45
South Eastern & Chatham Railway.	120 222	27 078	147 300	637.61	1 302.09	231.24	113.13	7 228 431	2.037	228 575	358.83
Isle of Wight Central . . .	195	293	488	28.47	29.70	17.6	16.43	62 550	0.78	1 583	55.6
Isle of Wight.	116	99	215	14.28	15.23	15.17	12.46	56 715	0.379	1 952	136.69
Plymouth Devonport & South Western Junction.	303	...	303	31.65	54.6	9.58	55.5	15 869	1.909	(Light Railway).	
Great Western Railway. .	463 361	149 098	612 459	3 702.59	6 187.15	165.83	98.99	31 102 975	1.96	929 865	251.15
North Eastern Railway . .	224 271	82 929	307 200	1 866.11	3 567.23	164.63	86.12	18 144 052	1.693	631 810	338.57
Great Northern Railway .	143 799	33 748	177 547	1 051.20	2 178.22	168.05	82.42	10 398 127	1.65	324 130	308.34
Great Eastern Railway . .	89 755	35 468	125 223	1 190.71	1 967.34	105.29	63.65	10 970 895	1.141	350 356	294.24
Great Central Railway . .	72 994	20 429	93 423	855.10	1 779.9	109.26	51.95	9 715 617	0.961	250 236	292.77
North British Railway . .	98 761	29 216	127 977	1 377.53	2 018.49	73.49	63.41	8 011 515	1.599	296 379	215.15
London & North Western Railway.	329 403	123 007	452 410	2 707.70	5 856.22	167.12	77.25	36 517 287	1.238	1 138 398	420.06
Furness	7 321	4 291	11 612	158.21	260.46	73.49	44.55	839 279	1.383	27 311	172.63
Midland.	209 206	63 078	272 284	2 169.20	4 099.62	125.5	66.42	23 114 278	1.177	738 516	340.48
Glasgow & South Western Railway.	33 840	13 214	47 054	493.36	857.32	95.44	54.95	3 210 233	1.465	118 012	239.22
North Staffordshire Ry . .	18 597	11 251	29 848	220.45	398.45	135.20	74.74	1 776 551	1.674	69 025	313.10
Caledonian	84 796	36 915	121 711	1 114.47	1 846.59	109.26	64.85	7 384 736	1.648	302 018	270.99
Metropolitan District Ry .	21 635	3 388	25 023	27.49	58.31	801.12	377.75	1 218 826	1.892	14 422	524.6
Metropolitan Railway. . .	20 921	1 803	22 724	65.30	141.65	348.0	160.48	1 397 233	1.626	16 571	253.76
	2 117 798	730 813	2 848 611	19 187.83	35 424.34	148.45	80.41	185 698 343	1.534	5 835 931	304.147

3° Working junctions at a distance from the station, or other signal-box, or the three junctions of a triangle lay-out from one signal-box;

4° Providing automatic signalling over long lengths of line;

5° Automatic control of trains, or semi-automatic train control, or train stop, a branch in which signal engineers have taken a foremost position in the country;

6° Semi-automatic, *i. e.*, automatic signalling for straight running through a station, with the signal-box normally closed, but when any shunting has to be done the signal-box to be used, in much the same way as a ground signal-box;

7° Substituting track circuits for facing point lock bars;

8° Automatic block, *i. e.*, providing track-circuited sections with illuminated indicator diagrams in signal-boxes to act as block indicators, and electric locks, controlled by the point detectors and track circuits. Controlling the operation of the levers;

9° Electrical interlocking instead of mechanical locking. Electrical interlocking can be made quite as safe as mechanical. The conditions in a frame in a signal-box renders it as safe as locking ties and dogs. It is very much lighter for signalmen to work and can be constructed at less cost;

10° Increase the capacity of the railway lines. This can be done in many ways known to the signal engineer. For instance, *a*) by the provision of intermediate advanced starters. *b*) By properly spacing signals. *c*) By re-arrangements of the lengths of block sections. *d*) By improved systems of signalling. The present two-position semaphore is practically universal in this country, but there are 33 different shaped arms in use, and the positions of the arms representing the various functions are not uniform in this country. *e*) The provision of three-position signalling. *f*) The provision of speed signalling. *g*) The introduction of daylight signals the operation of which is practically instantaneous, and thus many seconds are saved in operation by their use. *h*) The quicker movement of traffic by introducing power signalling schemes for points and signals. *i*) Arranging for the power plant to work automatically during the night times and so doing away with skilled labour. *k*) By arranging for uniformity in signal aspects and shapes of signals, etc., and a uniform system of block working. *l*) Electrically-worked points and signals do away with rods, wires, trunking on the ground, which is then free from these obstructions, which are costly. It avoids the difficulties of expansion and contraction of rods, wires, etc.

It is evident, I think, that to get the best results from signalling the various questions concerned with the movement of traffic from 1° the traffic, 2° locomotive, and 3° signal engineer's points of view must be treated as a whole-scientifically from a railway economic point of view, and the conditions considered by an expert signal engineer, who should have before him, not only the cost of installation and maintenance of apparatus, but also the cost of operation, and the cost of delay to trains should be in his hand. It is all one business question, and the factors cannot economically be dealt with separately by men who are fully acquainted with only one of the separate factors.

If a signal engineer is asked to get out a scheme for signalling a station or layout and an estimate for the construction and maintenance of apparatus, without being told the operating cost, the amount of traffic that is expected, or the operating class the signal-box will be, obviously he is at a disadvantage, and is not in a position to give the most economical scheme because he has only the first cost before him, whereas if he had operating and traffic conditions fully before him he may have been able to

rearrange his scheme more economically, and to better advantage.

The question of the economics of signalling has not been put forward by the Institution practically at all. This is a very important factor in the practical working of railways, and most important to signal engineers if they are to put the greatest advantages of signalling before the management of railways.

Much as one must admire the signalling work that has been done, it must be considered that this has been applied to conditions applying in the past, *viz.*, locomotive and traffic density. The acceleration of trains, their deceleration due to braking power, the speed of trains and the loads, were quite different to those of the present day. The greater acceleration of electric trains and increased deceleration due to greater braking power, and to electric traction, have, of themselves created quite a different set of conditions as regards signalling. The greater density of traffic demands greater speed, and more trains at frequent intervals. The electrical engineer has done his part as to speed, the locomotive engineer has given more powerful engines to draw heavier loads, the carriage and wagon manufacturers have provided larger wagons, the civil engineer stronger roads, and it remains for the signal engineer, although he has done much, to do his part in increasing the capacity of the lines by safe and efficient means.

Now the electrical and mechanical engineer, has perforce, to revolutionise his previous practice, and the signal engineer, in studying his problems, will have to look into the signalling of trains under the new conditions that now apply, and if he sees that the conditions demand new methods of signalling, it will be for him to face the problem boldly and propound new methods which, at first, may not appeal to all departments, but it will be his *forte* to bring the new methods forward.

The question of « Standards » is a very important and pressing one, and will have to be faced and solved. The question now should be of the standards that will meet the requirements for at least ten, if not twenty, years hence.

The economics of signalling includes the question of the operation of the signalling apparatus. This is essentially, in the initial stage at least, a signal engineer's question. Given the amount of traffic at a station or length of line, he will consider the various factors and be able to decide whether, under the circumstances, it will be more economical to provide manual operation by signalmen, or semi-automatic, manual by shunters, or automatic signalling. He will be able to decide the most efficient and economical working of sidings.

In dealing with a lay-out at a station where the traffic is dense, time of operation is of importance. It may be thought that because, in mechanical signalling, directly you pull the lever the points act, that this is as quick as it is possible to be, but this is not so in practice. It takes longer to pull a lever than it does a switch or press a plunger. Then sometimes two or three lever movements are required whereas, in power signalling, the various functions can be operated by one lever. Whether the cost of the time saved justifies any additional expenditure has to be considered by the signal engineer. With light signals, for instance, the change from one indication to another can be practically instantaneous, thus saving some seconds. Seconds at busy places are very important in maintaining the flow of traffic.

In the lay-out of running lines, stations and yards, the experience of the signal engineer can often offer valuable advice. The operation of a yard or station has to be done with due regard to the traffic on the main running lines, so that even if it is only a yard, or a siding, he can arrange the signals and the curvature of the cross-over giving access to the yard

correctly for the maximum speed of working.

To the younger members of the Institution I wish to offer a word of encouragement. Do not think all the problems of railway signalling and telegraphy have been solved. There are many waiting to be solved in the light of the past experience of the profession. This Institution, I think, will offer great advantages, and be more useful in the future to its members than in the past, owing to its getting on to the age of maturity. Do not, by any means, think that signalling will not offer you opportunities in the future. There are many openings and much scope for young men of good education, who will apply themselves to the study of the principles of the profession.

I could not conclude this address without acknowledging the good work done by the firms who have specialised in railway signalling and railway telegraphy, and who have contributed to the success of railway signalling and railway telegraphs in this country, and carried the work and practice to the Colonies and foreign countries, not only English practice, but the more modern principles of signalling that have come into force in some other countries. We trust those of to-day will be as successful, even more so, than their predecessors in helping forward the science and art of railway signalling and telegraphy.

MISCELLANEOUS INFORMATION

[625 .14 (01 (.73)]

1. — Otheographic records of wheel effects on rails.

(*Engineering News-Record.*)

Figs. 1 to 5, pp. 483 and 485.

In trial runs of electric and steam locomotives and motor cars, to test their balance and riding properties, it is important to determine the conditions of vertical load and lateral thrust exerted by the wheels on the track rails. Lateral thrust is not confined to curves, in the guiding of the engine by the wheel flanges, but may be experienced on tangent track due to a nosing movement or lateral sway of the engine. Besides the value of records of individual engines, the comparison of different types of engines as to these effects may prove of great practical value. To provide for determining and recording the load and thrust of engines in motion, a recording device known as the otheograph has been developed by the engineers of the General Electric Co. and installed in the company's 4 1/2 mile testing track at Erie, Pa. (fig. 1). This device gives a chart showing for each wheel of the engine both the vertical load and lateral thrust.

Fig. 1. — Otheograph installation at Erie, Pa.

The otheograph. — This recording instrument (fig. 2), consists of a heavy and rigid cast steel tie with T-shaped ends, each end having two bearings 12 inches apart for the base of the rail and two similar bearings against the outside of the rail head. The vertical and horizontal deflections of the rail between these bearings operate the recording apparatus by means of springs mounted in the ends of the tie. The point of contact for the vertical deflections is under the outer side of the rail head and not under the base. For the vertical loads, springs which give 1/8-inch. deflection for each 25 000 lb. of axle load are used with the lighter engines, or 1/8 inch for each 50 000 lb. with engines of heavier loading. In each case the maximum deflection is 3/8 inch. For the horizontal thrust, the springs have a deflection of 1/8 inch for each 20 000 lb. of pressure. The movements of the springs under passing wheel loads are recorded through a lever having arms of 8 to 1 ratio with a pointer at one end to trace a record on the paper chart on a revolving cylinder.

In the test track there are twenty-five of these otheograph ties, spaced 24 inches centre to centre and covering about 50 feet of track, including two rail joints. Wood ties bedded in ballast in the usual way carry two light wooden stringers upon which rest the ends of the steel ties, these stringers being used to maintain the alignment but having no appreciable stiffness. In this installation all the test ties are on tangent track, but they can be used equally well on curves.

The mechanism for revolving the charts or recording cylinders is driven from a constant-speed electric motor, with suitable gear reduction, so that the distances between the peaks on the curves are proportional to the wheel spacing on the various locomotives. Since all the cylinders are operated simultaneously they provide a series of records on successive ties for each wheel on each side of the engine. The speed of revolution is independent of the speed of the passing engine, so that records may be taken with any desired ratio between the ordinates and the abscissae. Diagrams taken in this way on any one tie show the amplitude and characteristics of both the load and thrust of all the wheels passing that tie.

The otheograms. — Typical diagrams or otheograms are shown in figures 3, 4 and 5, with the engines plotted upon them to show the relation of the wheels to the peaks or curves of the charts. The vertical deflection of the rail under load is recorded by the bottom line, while the lateral deflection is shown at the top. All lateral thrust of the wheels outward from the centre of the track is indicated by the record below the base line of the upper curve. It will be noted that at places this curve crosses the base or zero line, indicating a negative lateral movement. This is due to stressing the rail when the locomotive wheel is shifted on the rail toward the centre of the track. Reference to the marks above the zero line serves as an index to the lateral movement on the corresponding otheogram taken at the opposite end of the tie. The amplitude of the records shows a multiplication of the rail movement of approximately 8 to 1. The amplitude or height of the record indicating vertical pressure is approximately 58 000 lb. per inch, while that of the curve showing lateral movement is approximately 32 000 lb. per inch.

In studying the otheograms for practical purposes, the record from a slowly moving locomotive, would show the equalized distribution of the weight and such a record would serve as the basis for comparison with a record taken at high speed. The effect of side thrust in changing the vertical component, as well as any variations due to dynamic unbalance, would be quite noticeable. The effect of a wheel with a flat spot would also show very clearly. The record is not necessarily limited to one locomotive, for by moving the paper slowly the record of all the wheels of an entire train might be taken.

It is explained that the deflections shown in the upper curve, indicating lateral movement, cannot be considered quantitatively when above the base line. These negative indications simply show that the flange on the opposite side of the track is being pressed against the rail. For example, in the record shown at the right in figure 3 the leading truck of the steam locomotive indicates a lateral blow

Fig. 2. — Steel test tie fitted with otheograph.

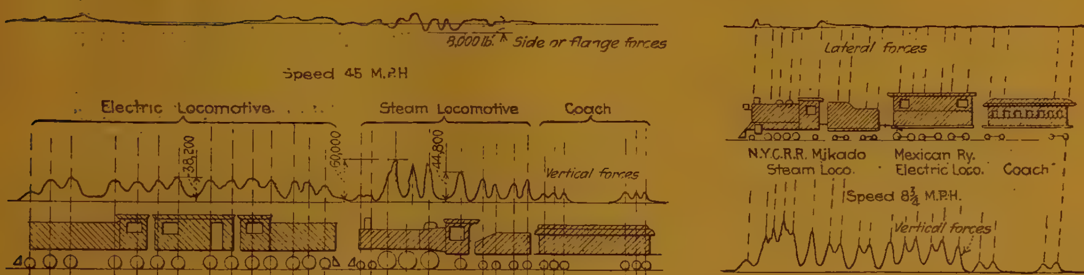

Fig. 3. — Diagram of loads and lateral thrusts of engine wheels.

Left: typical records from electric and steam locomotives and coach.
Right: record from steam locomotive hauling electric locomotive and passenger coach.

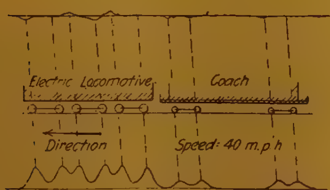

Fig. 4.

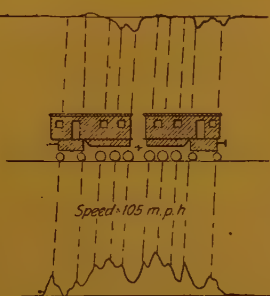

Fig. 5.

Figs. 4-5. — Diagrams
from electric locomotives.

Left: for Mexican Railway. — **Right**: for Paris-Orleans Railway.

against the south rail. The record on the opposite end of this same tie would probably show a positive indication of lateral deflection. Due to slight inaccuracies in the recording stylus, it is necessary to take into account the starting point of the base line at the left of the otheogram. This accounts for the slightly diagonal pitch of the dotted ordinates which have been plotted on the diagram between the axles and the respective curves. Having given the outlines of the locomotives under test it is feasible to lay out a diagram showing the several wheels with reference to the impact indicated on the diagram.

In the test indicated at the right in figure 3, the steam locomotive was hauling a new electric locomotive built for the Mexican Railway, this latter engine regenerating during the run at a speed of 8 3/4 miles per hour. The electric locomotive had three two-axle trucks with 46-inch wheels and 50 000-lb. axle loads. It will be noticed that the spacing between the trucks is less than the truck spacing; that is, the truck wheelbase is 110 inches while the distance between the axles of adjacent trucks is only 78 inches. The motors had pinions at each end, engaging gears on a driving shaft; this gearing was of the flexible type, designed to take up any slight variation in the machining of gears and pinions. The steam locomotive of the New York Central Railroad was of the 2:8:2 type, fitted with a booster engine on the trailing axle. It had 63-inch driving wheels with 66-inch axle spacing and 60 000-lb. axle loads. The leading axle is 9 ft. 8 in. ahead and the trailing axle 10 ft. 10 in. behind the adjacent driving axle.

Two diagrams from electric locomotives are shown in figures 4 and 5. The former is from the Mexican Railway electric locomotive hauling a coach at 40 miles per hour. It will be noted that, as might be expected, there is greater lateral thrust with the engine running in this way than when being hauled by another locomotive (as in figure 3), especially as the speed is much higher in the latter case. The diagram in figure 5 is from an electric locomotive built for the Paris-Orleans Railway, France, and running at the high rate of 105 miles per hour. This last engine has two six-wheel groups of driving wheels, with a four-wheel truck at each end. In this diagram lateral thrust is indicated by the second and third driving wheels and the two wheels of the rear truck.

[621 .134.1 (.45)]

2. — A multiple purpose automatic valve for locomotive cylinders.

Figs. 6 to 8, p. 487.

The valve, of which we give a description taken from the *Rivista tecnica delle ferrovie italiane*, serves the purpose of air valve, cylinder relief valve and bye-pass valve for equalising the pressure on the two sides of the piston when the engine is running with the regulator closed.

Figure 6 gives a sectional view of the apparatus now in use on about 800 locomotives on the Italian railways, and figure 8 shows the arrangement.

Each cylinder is provided with two similar valves fixed at each extremity and connected by means of a pipe H. These two valves are also provided with a pipe which leads from the steam chest and connects to the lower portion. The valve itself is double seated, the upper seat serving to make or cut off communication between the cylinder and the pipe H, while the lower seat, which is of larger diameter, separates the space in which steam chest pressure exists from the annular space which is in communication with the pipe H. The socket into which the latter is fixed is also provided with ports leading to the atmosphere.

It will be seen that when the regulator is opened, the valve is lifted and cuts off com-

munication with the pipe H and with the atmosphere. If an excessive pressure occurs in the cylinder, the valve will be forced down off its seat and the contents of the cylinder will escape through the pipe H and by the ports leading to the atmosphere. The pressure at which the valve acts as relief valve depends upon the ratio of the areas subjected to steam chest pressure and to the pressure existing in the cylinder.

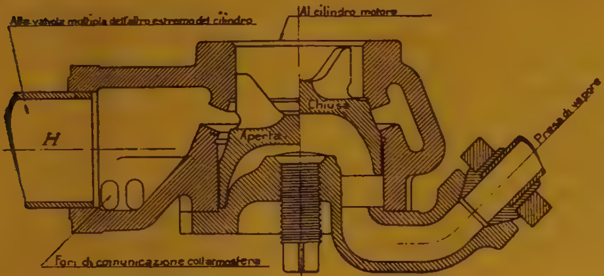

Fig. 6.

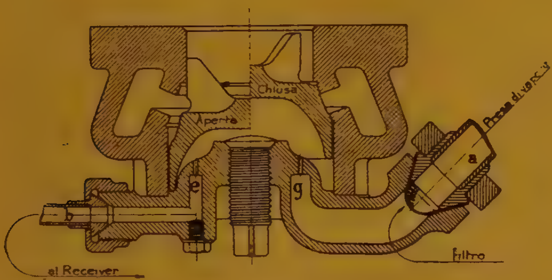

Fig. 7.

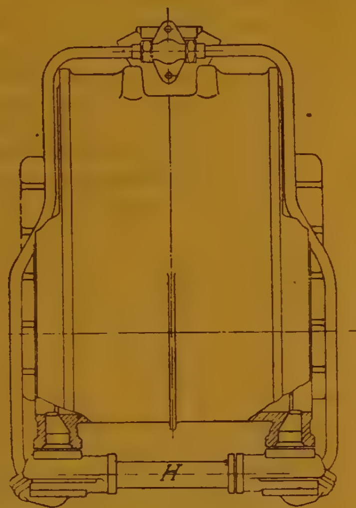

Fig. 8.

Figs. 6 to 8.

Explanation of Italian terms: Al cilindro motore = To cylinder. — Alla valvola multipla dell'altro estremo del cilindro = To valve at other end of cylinder. — Fori di comunicazione coll'atmosfera = Ports communicating with atmosphere. — Aperta = Open. — Chiusa = Shut. — Presa di vapore = From steam chest. — Filtro = Gauze strainer. — Al receiver = From receiver.

When the regulator is shut, the two ends of the cylinder are connected with the atmosphere through the ports in the body of the fitting, and also are put into communication with each other by means of the pipe H. The valve thus performs a function of an air valve, but the suction produced by the piston not only draws in air from the atmosphere, but also air from the other end of the cylinder. For this reason the cylinders are not cooled down to the same extent, and less dust is drawn in from the atmosphere.

The size of the pipe leading from the steam chest to the underside of the valve should be sufficiently large to ensure the valve closing rapidly when the regulator is open.

On compound locomotives originally the underside of the valve, in the case of low pressure cylinders, was connected to the receiver, which in this case is analogous to the boiler in the case of high pressure cylinders, but it was found that at high speeds, probably on account of high compression or considerable variations of pressure in the receiver the valve lifted and fell at each stroke of the piston. In order to obtain satisfactory

results, another arrangement was adopted with the object of maintaining on the underside of the valve a pressure intermediate between that in the high pressure steam chest and in the receiver, sufficient to keep the valve closed on its seat, but also sufficiently low to allow the valve to operate as a relief valve. This arrangement is shown in figure 7.

The space between the valve and the lower cover communicates by means of pipes *a* and *b* with the steam chest and with the receiver. There is therefore a current of steam passing through this space from the steam chest to the receiver, the amount being dependent on the area of the cross section of the orifices *e* and *g*.

The pressure existing on the underside of the valve depends on the relative size of these orifices. It is, however, necessary that the flow of steam shall not be so great as to cause appreciable loss. The dimensions of the orifices *e* and *g* have been fixed experimentally. It has been found that for a four cylinder compound locomotive good results are obtained if *e* is 2.5 mm. (3/32 inch) in diameter and *g* 3.5 mm. (9/64 inch) in diameter.

A very slight alteration in these orifices has a considerable effect. In order to avoid these becoming choked up by means of foreign matter carried over by the steam, a gauze strainer is placed in the connection with the pipe *a* which leads from the steam chest.

[385. (09.2]

OBITUARY

John E. FAIRBANKS,

General Secretary and Treasurer of the American Railway Association;
Member of the Local Organising Commission of the Session of Washington (1905):
Delegate at the Session of Rome (1922) of the International Railway Congress.

With profound regret we have received news of the death of Mr. John E. Fairbanks, General Secretary and Treasurer of the American Railway Association.

Mr. Fairbanks was a member of the Local Commission of the Congress at Washington in 1905, in the organisation of which he took an active part. Since that time he never ceased to take the keenest interest in the work of our Association.

The office of the Secretary of the American Railway Association has always been the chief point of contact, for American Railroad officers, with the International Railway Congress Association, and Mr. Fairbanks served as a delegate of the American Railway Association at the International Congress in Rome, Italy, in 1922.

Mr. Fairbanks was born in Jersey City on 5 December 1870, and was educated in that city. He entered the service of the American Railway Association on 21 April 1892, and on 1 June 1909, was appointed assistant general secretary and assistant treasurer; and was appointed general secretary and treasurer on 17 November 1915. While holding both of these offices, he held similar positions in the Bureau of Explosives, and was secretary of the General Managers' Association of New York. He also held similar titles in the Committee on Railway Mail Pay, and he had been the clerk of the American Railway Guild for the past twenty-five years. On taking the secretaryship, in 1915, Mr. Fairbanks succeeded William F. Allen, the well-known first secretary of the American Railway Association, to whom Fairbanks had been an able assistant for over 20 years. In 1915, the association did its work through 16 committees, the secretary being the guiding spirit in most of the work of the committees; but four years later, and following changes incident to government operation of the railroads, the scope of the American Railway Association was greatly enlarged, and the general secretary's activities now require offices in three cities, New York, Chicago and Washington. The expansion of the association beginning in 1919 has amalgamated with it no less than twelve very important associations of railroad officers formerly independent. The parent association at the present time consists of eight divisions, each with its own secretary, and nine sections; with approximately two hundred standing committees.

We offer to the family of our lamented colleague our sincere condolence and our deep sympathy with them in their loss.

The Executive Committee.

Dr. P. H. DUDLEY,

Consulting engineer to the New York Central Railroad;

Reporter to the International Railway Congress at the Paris (1900) and Washington (1905) Sessions, and delegate to the London (1895) and Berne (1910) Sessions.

We have heard with deep regret of the death of Dr. P. H. Dudley, which took place at New York on the 25 February last.

Dr. Dudley was one of our colleagues known by sight to most. His very special knowledge of rail material led to him being chosen to deal with this subject at the Paris session (1900). He dealt with the question « Nature of the metal for rails » in a remarkably able manner, as he did at Washington in 1905 with the subject of « Rails for lines with fast trains ».

Dr. Plimon Henry Dudley was born at Freedom, Ohio, 21 May 1843. He was therefore almost 81 years of age at the time of his death, which occurred at the Commodore hotel, New York city, Monday evening, 25 February 1924. His father was a well-educated and prosperous farmer who had emigrated from Massachusetts to Ohio. Young Dudley spent the early years of his life on the farm, and at the age of 20 entered Hiram college. He attended college during the winter months but returned home to help on the farm during the spring and summer. While at college one of the professors, a great admirer of Bessemer the English metallurgist, who in 1856 had invented the process for making steel which bears his name and which revolutionized the steel industry, had considerable to say on this subject. Mr. Dudley's interest was so aroused that he determined to take up the study of metals and metallurgy as his life work. He was so keen and industrious a student in science and engineeering that upon graduation he was able to secure the position of city engineer at Akron, Ohio, and held this office from 1868 to 1872.

He then became chief engineer of the Valley Railroad, running from Cleveland to the coal fields of Ohio. He held this position for two years, and during that period invented the dynamometer for measuring the drawbar pull of a locomotive. He gave up his position in order to devote his entire time to the perfection of this instrument. In 1876 he had added a recording device, and the results he obtained together with his ability to analyze the data he secured, attracted the attention of the Eastern Railway Association, and they engaged his services in 1878. The dynamometer gave a scientific and convincing demonstration of the inefficiency of the locomotive as it then existed.

In 1880 he became associated with the New York Central & Hudson River Railroad and was in continuous service with the New York Central Lines until the date of his death. He first advanced the theory that the rail acts as a girder. This theory met with much opposition and ridicule and as a result of this controversy, he designed a track indicator which recorded irregularities of surface, level and gauge, and which is in common use throughout the country in inspection cars. He also invented the stremmatograph to determine the manner in which the load of a moving train is transmitted to the track support. The railroad constructed for him a special car in which were housed the dynamometer and inspection equipment. There was also a laboratory and workroom as well as living quarters. In this car Dr. Dudley and his wife lived for 33 years and this was their only home until her death about two years ago. He afterwards continued to live in the car for

nearly two years. A short time before his death he moved to the Commodore hotel, where he lived until he passed away.

Dr. Dudley proved by the records taken from the track indicator that his theory of girder action was correct, and designed a rail which gave the results he predicted. The rail he designed was five inches in height, and although he convinced the management by his arguments, they very cautiously purchased only five miles of rail for test purposes. This rail was placed between the Grand Central station and Mott Haven. It was difficult to believe that a rail five inches in height would remain stable and not overturn under a locomotive or loaded train. When it was found that the rail did not tip over as expected, a considerable tonnage was ordered, as the riding qualities of the track were greatly improved and the cost of maintenance decreased. Later he was permitted to add one-eighth inch to the height, and gaining confidence in his judgment, 90-pound rails 5 1/2 inches high were eventually permitted. These events mark the adoption of heavier and better designed rail sections by the railroads of the country.

Although his inventions and discoveries covered a wide field, the major portion of his life was spent in an effort to improve the quality and manufacturing methods of the steel which goes into rail. He worked from the standpoint of a true scientist, with no thought of personal gain. He did not patent any of his inventions or discoveries, but gave them out for the benefit of all who choose to use them.

In a message notifying the officers and employees of the New York Central Lines of Dr. Dudley's death, the late president A. H. Smith said :

« By his extraordinary ability, broad scientific knowledge and enthusiastic devotion to railroad service for an unbroken period of forty-eight years, Dr. Dudley contributed at least as much as any other man to the development of American railroads. Perhaps it would not be an overstatement to say that he did more than any other one man to confer upon humanity the boon of rapid, safe and economical transportation. His life work added lustre to the fame of the New York Central. His career was characterized by arduous toil in the search after scientific truth, tireless energy and unimpeachable loyalty and integrity. It was Dr. Dudley's scientific development of the track which made possible the first 100-ton locomotive, an epoch-making event. Dr. Dudley was a metallurgist as well as an engineer. His studies resulted in the manufacture of steel rails which greatly reduced the number of accidents due to broken rails and led to great economies. »

Dr. Dudley was an honored member of many engineering and scientific societies, among them the American Railway Engineering Association. At the time of his death and for many years he was a member of the rail committee of the association.

His funeral, which took place at the Church of Heavenly Rest, New York city, Thursday, 28 February, was attended by many prominent railway officers, and representatives of engineering and scientific societies, of which he was a member. The honorary pallbearers were : A. H. Smith, president, New York Central lines; H. P. Whitlock, New York Academy of Sciences; J. B. W. Reynders, American Institute of Mining & Metallurgical Engineers; George J. Ray, vice-president American Railway Engineering Association, and George Pegram, American Society of Civil Engineers.

We offer to the family of our lamented colleague our sincere condolence, and wish to express the sincere sympathy we feel with them at their loss.

The Executive Committee.

NEW BOOKS AND PUBLICATIONS

[621 .137.1 & 385. (04]

HALLEUX (F.), inspector of traction and rolling stock on the Belgian National Light Railways. — **Manuel du mécanicien des chemins de fer vicinaux et d'intérêt local (Manual for drivers of light railways and local lines)**, 2nd edition. — One volume in 8vo (7 3/4 × 5 3/4 inches) of 782 pages, with 141 figures. — 1923, Publishers : Ramlot Brothers & Sister, 25, rue Grétry, Brussels, and H. Dunod and E. Pinat, 49, quai des Grands-Augustins, Paris. — Price : 25 francs.

Judging from the title Mr. Halleux has given this book, it seems quite natural that it should contain 782 pages, but the author informs us in his preface that this second edition, on account of the important additions it contains, practically makes it a new work, intended to be useful, not only to drivers and firemen, but to workmen, inspectors and others connected with the working of light railways and local lines.

After going thoroughly into the matter, as far as the enginemen are concerned, he has reserved a large portion of the book for dealing with the subject of upkeep and repairs which are effected in the shops or sheds. The reader who studies this work with sufficient interest will acquire a very comprehensive knowledge of the locomotive, become familiar with the various circumstances in which it is used, and then learn all that is necessary to keep it in proper working order.

The book is divided into thirteen chapters which may be grouped under the three following headings : Description of the locomotive; the locomotive when running (working and driving); upkeep and repairs.

The first of the two introductory chapters deals with the historical aspect, and the second with preliminary ideas relating to the subject. At the beginning of such a book it is proper that we should be reminded of the debt we owe to the pioneers of steam traction and of the many efforts for improvements that have continually been made from Cugnot's trolley up to the engine *Le Belge* which inaugurated the construction of locomotives in Belgium. The second chapter is useful in that it gives the reader an elementary knowledge in physics and mechanics; it explains the properties of steam, the phenomena relating to combustion, the qualities of fuel in general use, and gives an account of the materials used in locomotive construction.

Chapters III to VII give a description of the locomotive itself, beginning with a general outline in which the principal parts are described, shewing their construction, the position they occupy and their use. Each component is then discussed and examined in detail. In chapter IV the author points out the chief characteristics of locomotives such as tank engines which run in either direction, and describes every type of engine in use on the Belgian light railways and gives their principal dimensions in a table. The three following chapters deal with the boiler, the mechanism and the framing. Every part of the boiler is carefully explained, as well as all the fittings, such as the feed, safety, superheating, exhaust appliances, etc. The important question of the quality of waters and boiler compositions is also gone

into. The points specially dealt with in the chapter on mechanism are : the work accomplished by the steam in the cylinders, the working of the valve gears by means of links, lubricants and lubricators, packings and special valve gears.

Chapter VIII gives a description of the principal parts of compressed air and vacuum brakes, with an explanation of their working.

Chapter IX gives an analysis of the various circumstances which influence the stability of locomotives.

In chapter X, which deals with the power and efficiency of locomotives, the author shows, with suitable examples, how the resistances to be overcome are estimated : resistances due to running, curves and inclines, and how the loading and fuel consumption are calculated.

The two following chapters are devoted to the locomotive as much as regards running on the road as in the running sheds. In them the author minutely describes all the operations in the shed that are necessary to prepare engines for running : examinations and cleaning, sweeping the tubes, washing out, firing up, and points out and explains the rules that should be followed in working locomotives, especially as regards superheated locomotives. He then passes in review the principal incidents which may happen on the road and the means to obviate any trouble that may arise from them.

The last chapter, which contains 184 pages and is entitled : *Upkeep and repairs*, could in itself be classed as a manual for the use of those who deal with repairs, as they will find in it methodical methods of dealing with all cases that have to be attended to in the running sheds. These include first of all general upkeep, then repairs properly so called relating to failures or to wear of the boiler, mechanism, frames and wheels. There are also to be found methods of an essentially practical character concerning special work, such as that appertaining to the smithy, tempering, case-hardening, modern methods of welding (oxy-hydrogen, oxy-acetylene, electric, alumino-thermic, etc.).

We have not been able in these few lines to give more than a rapid outline of the extensive subject dealt with by Mr. Halleux. For each point mentioned, and he has left few in the back ground, he has succeeded in condensing all the necessary theoretical knowledge, and present in an appropriate form all the information necessary to enable the staff connected with the locomotive service to do their work intelligently. His comprehensive work will render it unnecessary for them to resort to other sources for the greater part of the information they may require, and will always be worth referring to, not only by the staff of the light railways, but also by that of railways in general.

E. M.

[62. (01 & 385. (04]

NACHTERGAL (A.), engineer, technical school professor. — **Notes sur le calcul des moments d'inertie des fers profilés (Notes on the calculation of moments of inertia for iron sections).** — One volume (9 1/2 × 6 1/4 inches), of 18 pages, with 22 figures in the text. — 1923, Published by Alb. De Boeck, 265, rue Royale, Brussels. — Price : 3 francs.

The object of these notes is to show, by means of numerical examples, that one is able, in calculating the moment of inertia of iron sections, to neglect

round sections and consider them as being of rectangular shape.

The numerical examples used by the author show that the difference between the two methods of calculation is practically negligible.

J. V.

[621. (02 & 385. (04]

NACHTERGAL (A.), engineer, technical school professor. — **Notes et formules du contre maître mécanicien (Notes and formulæ for the foreman mechanic).** — One volume in 8vo (7 × 4 1/2 inches) of 350 pages, with 289 figures in the text. — 1920, Published by A. De Boeck, 265, rue Royale, Brussels, and Ch. Béranger, Publisher, 15, rue des Saints-Pères, Paris. — Price : 20 francs.

In this work, everything that should be known by a good mechanic is to be found.

Theoretical knowledge is fairly well distributed, too much so perhaps, but numerous practical examples make the work highly interesting.

Each subject dealt with is followed by practical illustrations, and this to the fullest extent.

The work, however, appears to us to be somewhat too advanced and comprehensive to appeal to the general run of mechanics.

Nevertheless, let us hope that a book of this scope will prove successful, for as the author says in his preface : « As in course of time tools of various kinds are improved and new methods of work are instituted, it is requisite that the theoretical knowledge of workmen should be increased more and more. It would be valueless for engineers to create new types of machines if a staff could not be found to work them economically. »

J. V.